GENERAL MOTORS

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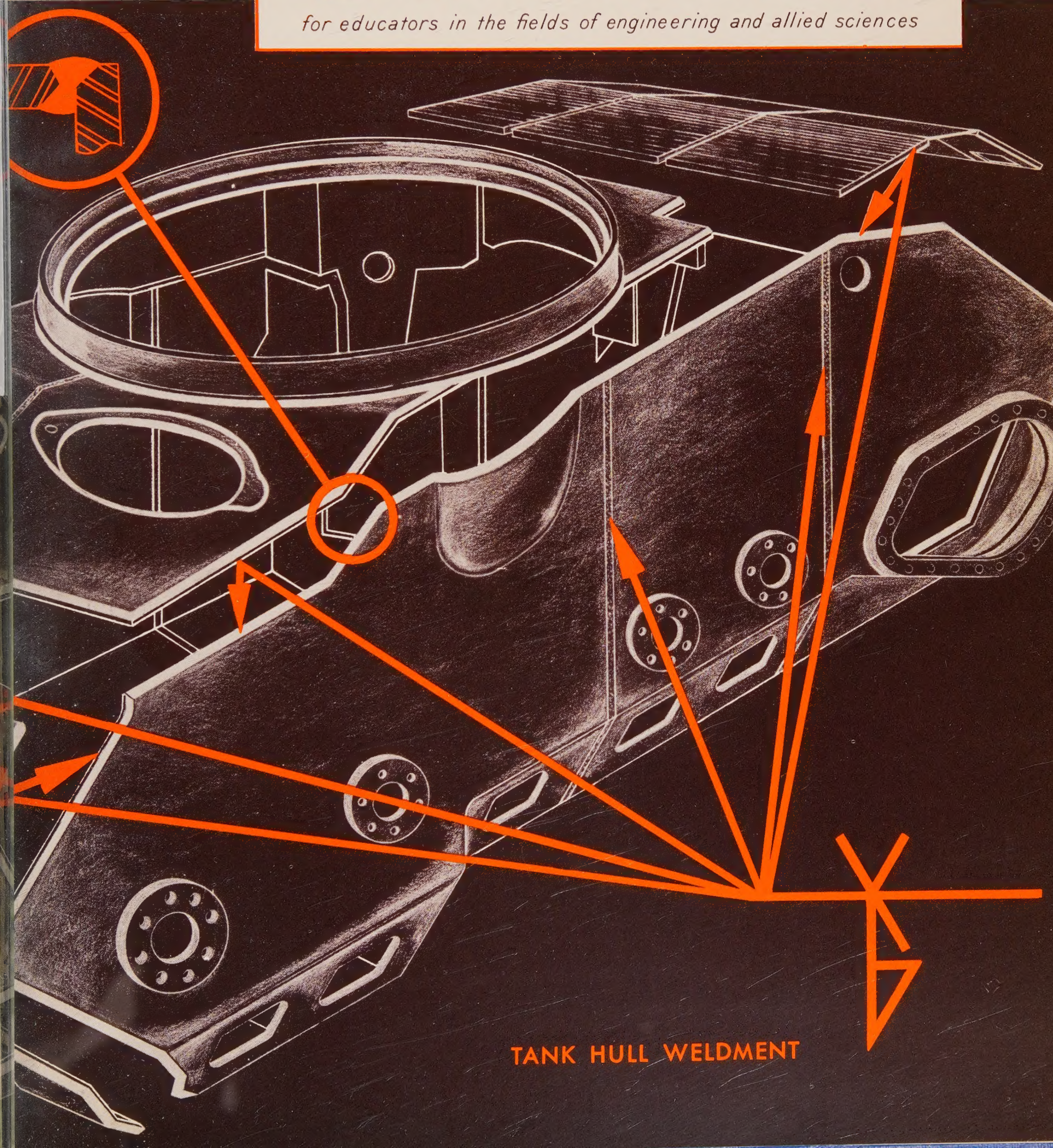
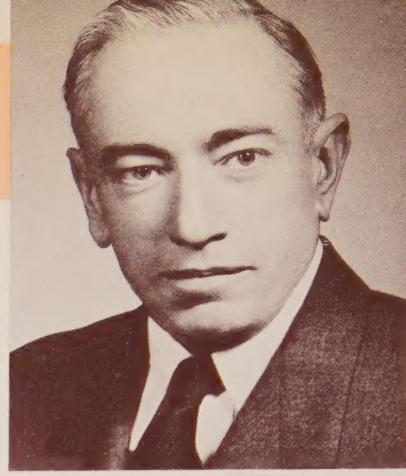
for educators in the fields of engineering and allied sciences



TANK HULL WELDMENT

On Management Abilities for the Engineer



To an increasing extent, today's engineer is becoming a manager. This is true at various levels of his work, whether he directs merely his own technical assignment, whether he directs a laboratory group, whether he directs an engineering department, or whether he enters the ranks of general management as many engineers do. But whatever the level, the engineer with management abilities usually gains maximum satisfaction from his performance and makes the greatest contribution to society.

Management always is concerned with problems. Fortunately, the engineer is singularly qualified in this respect for his basic education is designed to develop the ability to solve problems. Yet the art of managing is essentially a social skill. Thus, the engineer—to be a good manager—needs to include in his educational program the development of his own social skills.

Management has changed radically in the last two or three decades. The *boss* of yesterday has been replaced by the *manager* of today. The old-time boss felt secure in his conviction that he knew everything there was to know about his business and anything that could possibly affect it. The physical world around him held little that seemed likely to turn out to be an unknown quantity, either then or in the future. Unlike the boss of years ago, the manager of today does not hold himself aloof from the currents of thought of his time. He is keenly aware of what is happening in the world about him. He has a genuine interest in people. He communicates effectively with people. He is sensitive to public attitudes. He understands and respects economic principles. In short, management has taken on new and larger dimensions.

Looking ahead, the tasks that await future managers will be more complex than they are even today. Technology is accelerating at a pace never before equaled in the history of America. Horizons in all branches of knowledge are being pushed outward farther and farther. The automotive industry, for example, is certain to move forward to new heights in the years to come because of such factors as increasing population, suburban living, new highways, rising incomes, and a persistent customer demand for product improvement. Advances are coming down the road in other industries too: home building, commercial aviation, electronics, and power generation, to name a few.

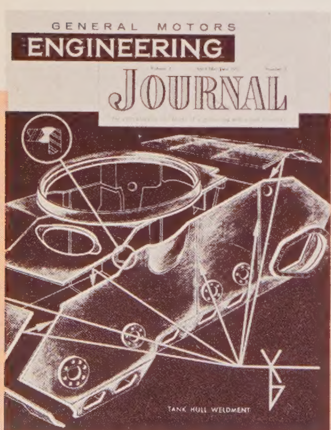
The manager of tomorrow has an outstanding opportunity to take part in and contribute to the revolutionary advances that are certain to take place. Fundamentally, his background and training should provide him the capacity—both as a manager and as a citizen—to assume leadership responsibility in maintaining and strengthening our free society. He will need to be a leader of thought, as well as of action. This is, perhaps, the newest and most significant dimension of the management job.

The challenge facing tomorrow's manager is leadership, not only in technical and administrative matters, but, just as importantly, leadership in the areas of human relationships and the development of social and economic patterns of our society. The engineer who desires management abilities can meet this challenge if he plans his education wisely so that he will be equipped to meet these broader demands. The *formal* education of the engineer properly concentrates on studies of the laws of natural science and their application. But the opportunity also is provided in his curriculum to direct him toward the development of the social skills of the manager. More importantly, the young graduate engineer must take it upon himself to continue this development beyond formal education after he joins the industrial world.

Engineering students now in colleges and universities will spend their careers in a period of unprecedented growth in the nation's economy and technology. They can look forward to increased reward and increased contribution to society if they have developed in themselves the art of managing.

Roger M. Kyes

Roger M. Kyes,
Vice President and
Group Executive



THE COVER

This issue's cover design—contributed by Herbert Combes of Cadillac Motor Car Division's Cleveland Ordnance Plant—symbolizes the application of modern welding techniques to join various armor steel-plate components of a military tank hull. It is necessary that careful consideration be given by the designer to the form of welded joint used in order that maximum strength and joint efficiency will be obtained with uniform stress distribution along the joint. Since these stresses are repetitive and at relatively high frequency, the metallurgy of the weld and development of stress rising notches must be watched closely to prevent fatigue failure.

The welding symbols represent the graphic method for conveying to the welding operator the designer's specifications regarding a welded joint, pictorially represented by the cross-sectional views.

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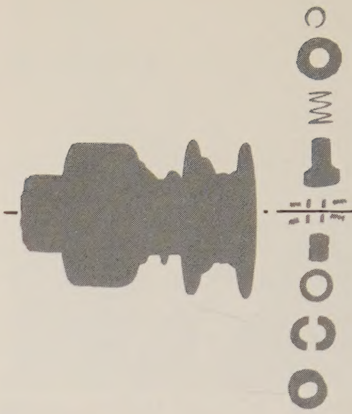
A Discussion of Design Factors for a 12-Volt, 4-Pole, Wave-Wound Automotive Engine Cranking Motor



The introduction of a 12-volt automotive electrical system on some 1953 General Motors cars was made necessary by the need for increased electrical system capacity occasioned by the use of higher compression engines. An important component of the 12-volt system was the engine cranking motor which was required to provide a faster cranking speed for the larger size engines. To meet this requirement, a comprehensive basic design program was undertaken by Delco-Remy Division engineers to develop a cranking motor having the required torque characteristics. Using basic engineering fundamentals of electric motor design along with experience gained from over 20 years in the design and manufacture of 6-volt cranking motors, product and manufacturing engineers worked in close cooperation to develop a cranking motor economical to produce and fulfilling all operating requirements.

THE automotive engine cranking motor is a special type of high-torque, d-c motor which operates under great overload to produce high horsepower for its size. The function of a cranking motor is to transform electrical energy from the battery into mechanical energy used to crank the engine.

Prior to 1953, 6-volt electrical systems met the requirements for ignition, generator output, and engine cranking speed. The advancing trend toward higher compression engines, however, required an increase in electrical system capacity which resulted in the introduction of 12-volt electrical systems. This introduction, in turn, called for a 12-volt cranking motor to which was added a requirement for faster engine cranking speed.

Torque Characteristics Were Basic Design Consideration

The change from a 6-volt to a 12-volt automotive electrical system made necessary the inauguration of a basic design program for a 12-volt cranking motor. The specific characteristics of the overall 12-volt system posed many problems in both the electrical and mechanical design of the motor.

The primary objective of the Delco-Remy Division design program was to develop a motor which would give required torque characteristics with minimum use of copper and iron and minimum current draw from the battery. Also, the matter of economical manufacture had to be considered when designing the individual components of the motor. This was achieved by close cooperation between engineering and manufacturing groups starting in the early design planning period (Fig. 1).

Because the overall program was of unusual magnitude, the final design was evolutionary in nature. In each new car model year succeeding the initial introduction of 12-volt cranking motors, electrical, mechanical, and metallurgical improvements were introduced as soon as exhaustive developmental tests indicated their adaptation was feasible. The end result is the compact, powerful, high-speed cranking motor, well protected from the elements, now being produced for all 1957 General Motors cars after "pilot" production in 1956.

Electrical Design Required Attention to Many Fundamentals

The first phase of the design and developmental program was concerned with determining the engine cranking requirements for the 12-volt cranking motor. Extensive cold room tests were undertaken in which starting torque requirements for the new engine designs were established. Also, batteries were tested under simulated cold weather conditions to provide design information. Cranking motor performance is so intimately connected to battery performance that both must be considered simultaneously.

After all basic and pertinent information was obtained, the electrical design

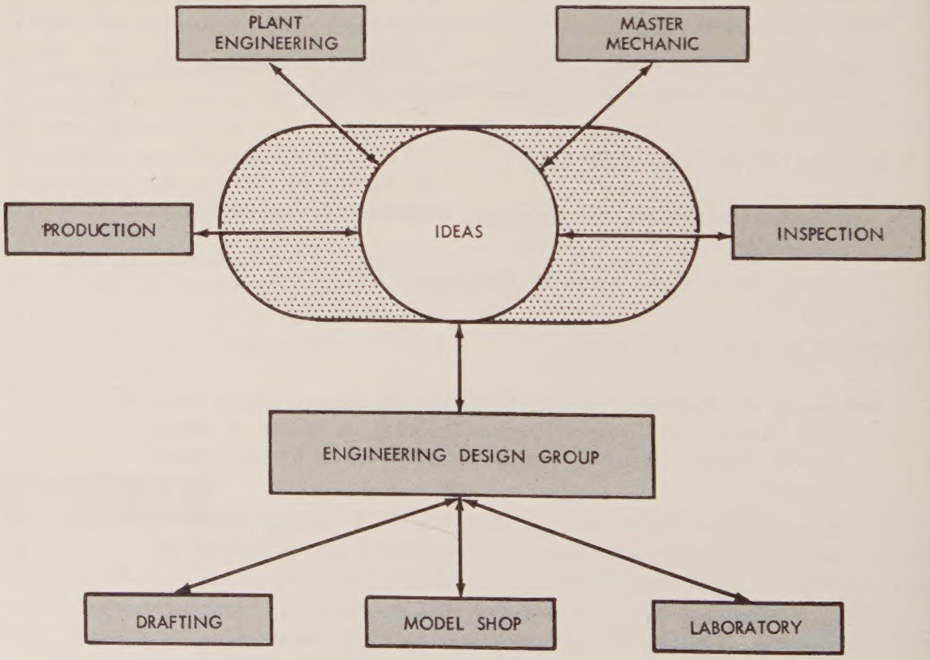


Fig. 1—This diagram illustrates the various 2-way roads taken for ideas, their sifting, and their execution during development of the 12-volt cranking motor.

By WILLIAM C. EDMUNDSON
Delco-Remy
Division

Electric motor design principles applied to meet engine cranking requirements

of the motor was undertaken. Attention to detail and tedious calculation of fine points were involved and were justified, considering the number of motors to be produced year after year. The following discussion describes in detail the several special approaches used to solve major problems connected with the electrical design of the motor. Most of the conventional calculations are assumed to be familiar and are not included in this discussion.

Frame Choke Zone

In a dynamo, with the axial length of the frame greater than ½ the perimeter of the pole body, the flux must pass from the pole to the frame through an entrance area less than either the effective frame area or the pole body area. This entrance area through which the flux must pass is called a "choke" (Fig. 2). The effect of this choke extends to a region equidistant from the pole body, whose perimeter area is equal to the effective frame area (or where the perimeter is twice the axial frame length). The length of the path is the distance between the pole perimeter and the region just described. The effective choke area is calculated at ⅓ of the path length from the pole edge and is equal to the perimeter at this region multiplied by the frame thickness.

This method for calculating the choke area is similar to calculating the tooth area of an armature having straight slots and wedge-shaped teeth. In this case, the tooth area is calculated at a radius of ⅓ the distance from the bottom of the tooth to the outside of the armature. This is a logical calculation, since more ampere-turns per in. are required in the part of the tooth where the area is restricted than are needed in the end of the tooth where the area is greater. It has been checked for accuracy by calculating the tooth path in small increments.

Pole-Tip Saturation

Automotive design practice requires minimum size and use of material for most components. In view of this, it became necessary to calculate carefully the pole-tip thickness needed. Excessive thickness would require the starting motor to have a diameter larger than necessary.

For intimate study of pole-tip saturation, the flux-tube method was extended beyond its original conception for air gap and teeth. The *flux-tube method* divides the series path through the air gap and teeth

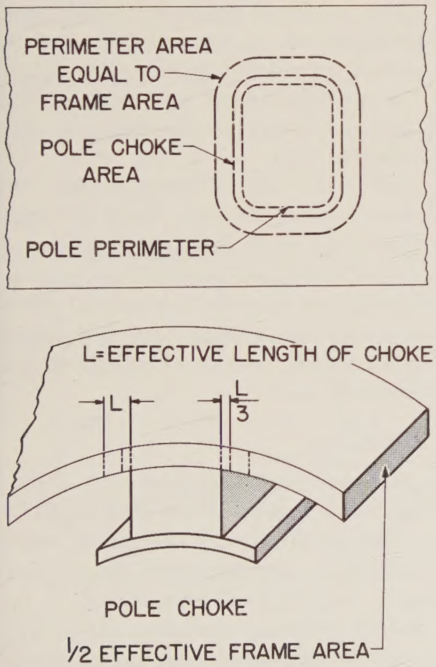


Fig. 2—The frame choke is a path expanding in area from the perimeter of the pole until its perimeter area equals the frame area. The choke exists when the length of the frame is sufficiently longer than the pole to have an area exceeding the frame thickness times the perimeter of the pole. The effective area is figured at ⅓ the distance from the pole to the outer perimeter.

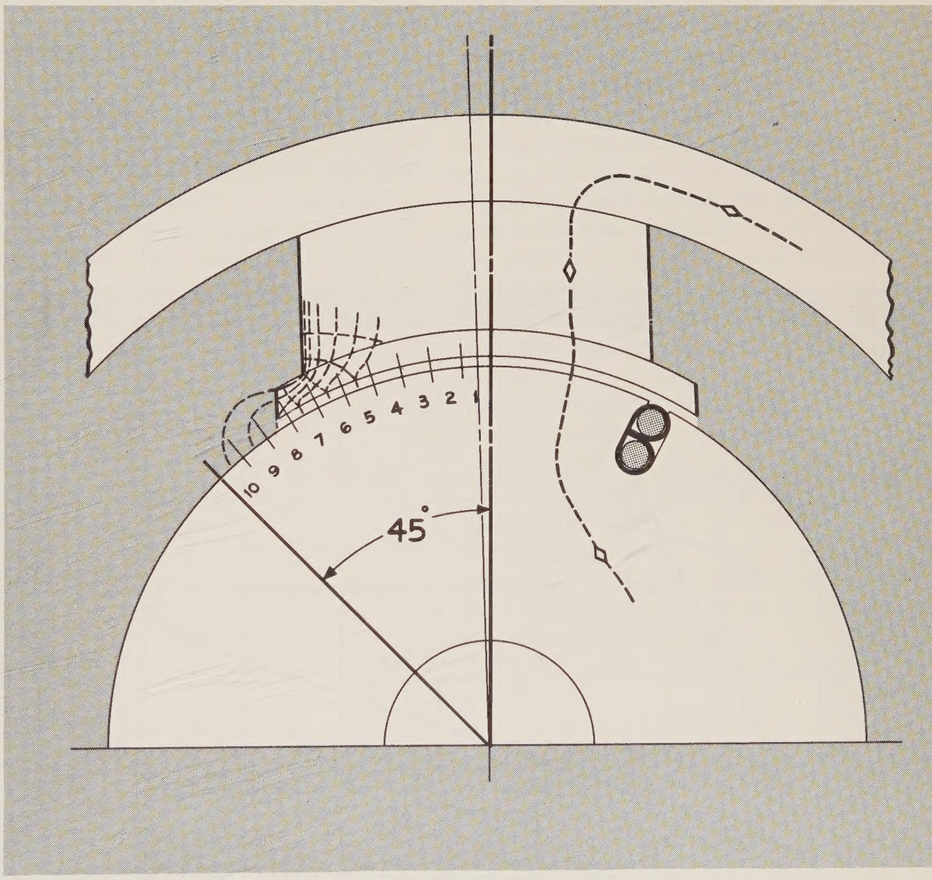


Fig. 3—The flux-tube method was used to study pole-tip saturation of the cranking motor. Flux tubes were equally divided along with parallel flux paths through the teeth and air gap and through the pole tip where applicable. In the varying air gap from the center of the pole to the neutral zone, such a fine division of paths was necessary to give accuracy in a calculation process involving approximations. This drawing shows, by dashed lines, the flux path traced through the pole tip and air gap of tubes 5 to 10. The dashed line in the right half of the drawing indicates the general flux path. The end points of the saturated pole-tip flux path were approximations based on calculations as to the point where flux density falls below 100,000 lines per sq in.

into numerous parallel paths, or "tubes," of equal area (Fig. 3). The areas are small enough so that the conditions at the center of each path may be considered as average and used as the basis for calculation of a saturation curve of the path. The divisions are small not only because the air gap is a variable from tube to tube, but because the magnetomotive force (mmf) is different for each tube, as a result of armature reaction. With a postulated net mmf for air gap, teeth, and pole tip and calculated apportionment of armature mmf, the total flux through these paths in parallel can be determined. The pole tip for the cranking motor was divided into tubes placed in series with the appropriate air gap and teeth tubes.

The calculation of pole-tip saturation has been the subject of much study, because the solution is neither direct nor obvious. The problem is changed by every variation in pole body width and pole face span. Much practice, however, has led to a simplified and practical approach, involving the following considerations and approximations:

- The thickness of the pole tip acts as a flux gate, and the exact length of the path is not important
- The area of the flux gate apportioned to each tube is inversely proportional to the air gap length (or reluctance of the air gap and tooth path) for each tube fed by the pole tip. This is necessary in order to have the same mmf drop across each tube in the pole tip
- A graphical presentation of the flux path in the pole tip usually indicates the gate area and the saturated length of path.

These approximations are justified when reference is made to saturation curves for air gap, teeth, and pole tip for one tube (Fig. 4). Such saturation curves indicate that the flux gate is of greater importance than the length of the saturated path in the pole tip. Also, a pole tip which is too thin almost excludes useful flux from the armature surface beyond the edge of the pole body.

Armature Slot and Conductor Design

The usual method for assembling conductors into slots involves first preforming the conductors into "hairpins" and then pushing them lengthwise, as a group, into the slots.

It would have been most desirable from the standpoint of performance to use rectangular conductors for the cranking motor, with the long dimension in the radial direction of the slot. With these proportions, however, the small conductors required for the 12-volt design were entirely unsuitable. The ribbon-like shape did not have the required stiffness for assembly, and the lamination slots were too narrow for punching. This shape also would have required high commutator risers, using an unnecessary amount of material.

Ordinarily, a square wire (Fig. 5a) would have been next most desirable. A square wire with rounded corners, however, could not be gripped well enough to be twisted accurately into the commutator risers. Round conductors were finally specified (Fig. 5b). The round conductors eliminated the need for twisting during assembly and reduced assembly friction to the point where the stiffness of the round section was acceptable.

The round conductors were well suited to adoption for cranking motor use. The shorter risers required by the round wire could be formed integrally on bars made by cold heading, thus eliminating the need for a 3-piece bar. With fewer pieces, the circumferential stack-up was reduced. Other sizing operations were added. Belleville-type, steel V rings were incorporated, and the core was tightened by hot-riveting. The resulting assembly was of more uniform quality.

The decision to use a partially closed slot further added to the advantages of the round conductor. Although the round conductor ordinarily would make inefficient use of space, in this case it worked out advantageously (Fig. 5c).

A partially closed slot was specified because calculations and checking tests indicated that: (a) a fully closed slot resulted in a loss of torque because the short-circuited slot flux displaced useful flux in the teeth, (b) a slot opening of 0.03 in. overcame this effect, and (c) any opening less than the 0.130-in. full width and more than 0.03 in. improved torque characteristics over that obtained with a full-width slot.

Laminations

It was not necessary to make the cranking motor laminations of thin stock for 2 reasons: (a) at very low speeds, when the motor operates under the greatest load, the iron loss is not enough to influence operating conditions and (b) any iron loss which does occur at high speeds is useful in limiting the free speed of the series motor, protecting windings and bearings. Accordingly, 0.060-in. thick material was chosen, being the least expensive steel still thin enough for punching into laminations.

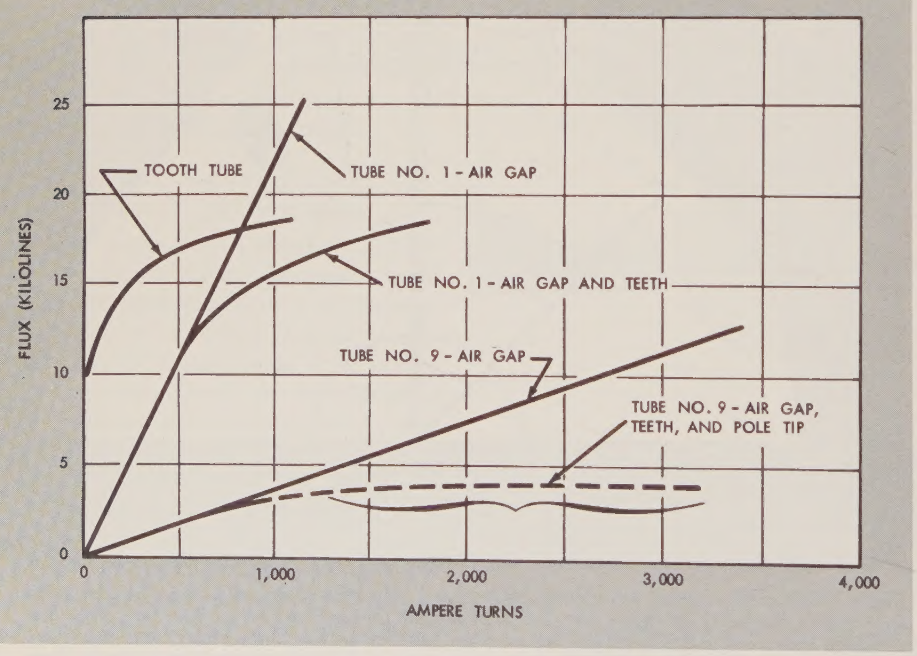
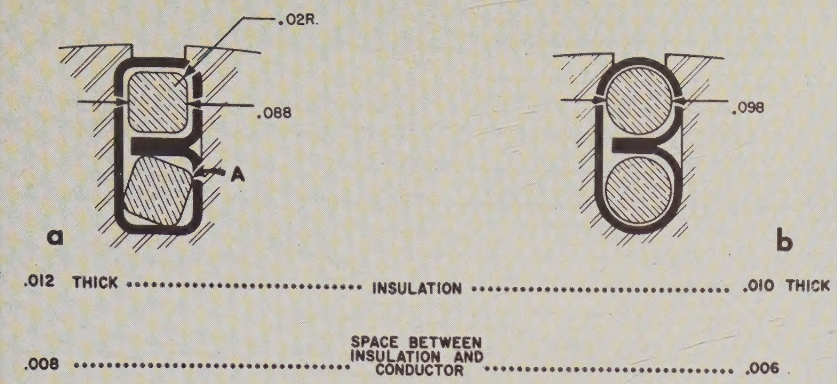


Fig. 4—This graph shows typical air gap, tooth, and pole-tip saturation curves. Tube No. 1 has no pole-tip saturation. In tube No. 9, the air gap is so long that the lamination teeth are not a factor. Other tubes have characteristics falling between these. The part of No. 9 marked off by the bracket shows the "flux gate" effect in the pole tip. The length of the path has little effect on the slope of this curve.



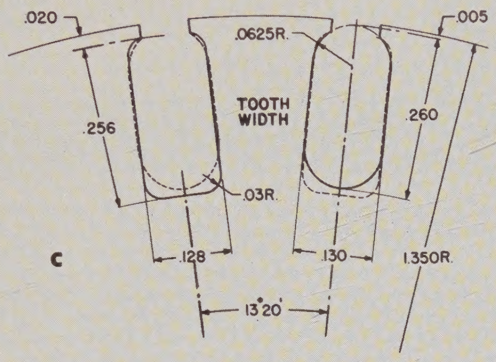


Fig. 5—Both square and round conductors were considered during the armature slot and conductor design stage. Analysis showed that square conductors (a) would have to fit loosely because the flat-bottomed slot and insulation bound all of the surface, and a slight twist from proper position would cause interference (*A*). Also, the square conductor would have to be totally below the overhanging lip. By using round conductors (b), with only a small area of friction, the conductors could be fitted more tightly. Without corners to tear the insulation, the insulation could be made slightly thinner. Part of the cylindrical edge of the conductor would come nearer to the armature surface and reduce the effective length of the tooth. Round conductors were specified for the cranking motor. With the round conductor the effective width of tooth is just as great, since it is at a larger diameter (c). Effective tooth length is actually less.

Load Saturation Curve

In general, the load saturation curve of a dynamo is not calculated directly. Direct calculations would involve the use of simultaneous equations of the saturation curves of the various series and parallel flux paths. The simultaneous equations, in turn, would be approximations.

To determine the load saturation curve, an estimated amount of ampere-turns were applied to the air gap, teeth, and pole tip. The flux was then calculated. From saturation curves for the series iron paths, the ampere-turns for this flux were read. This procedure was repeated until the arbitrarily chosen figure plus the latter figure equalled the total field ampere-turns available.

The torque curve for the 12-volt, 4-pole, wave-wound cranking motor was calculated directly from the flux determinations by the following formula:

$$T = 0.235\ (I\phi Z)\ 10^{-8}$$

where

T = torque (lb-ft)
I = motor current (amp)
ϕ = flux per pole
Z = number of armature conductors.

Running torque was not calculated by the usual design method of subtracting the effect of losses. In the operating speed range of a cranking motor, windage, friction, and iron losses are small and difficult to determine. At best these losses are determined either empirically or by tests on a similar motor. A more direct empirical method is to subtract a constant amount, which is determined from previous experience on similar motors, from the calculated torque curve. The amount is substantially constant because, as the bearing friction increases with higher torque, the iron loss decreases with lower speed (Fig. 6).

Compound Fields

A special flux path calculation for the cranking motor was imposed by the 12-volt design. In order to limit the free speed of this series motor, for quietness of operation and longer wear, shunt windings were required. The least expensive way to compound the motor was to place the shunt field on one pole and series fields on the other 3. This posed an obvious design problem, since the fields were equal at only one value of current and voltage. Calculations based on small increments of the flux path gave 2 important answers:

(a) Since the flux paths were highly saturated through the useful range, the flux division acted as if the individual fields were totalled and averaged. This gave an even flux distribution which helped to keep the air-gap pulls balanced

(b) Although the effect of the shunt field was reduced as current draw increased, since the battery voltage drops with current output, the torque curve was not affected ad-

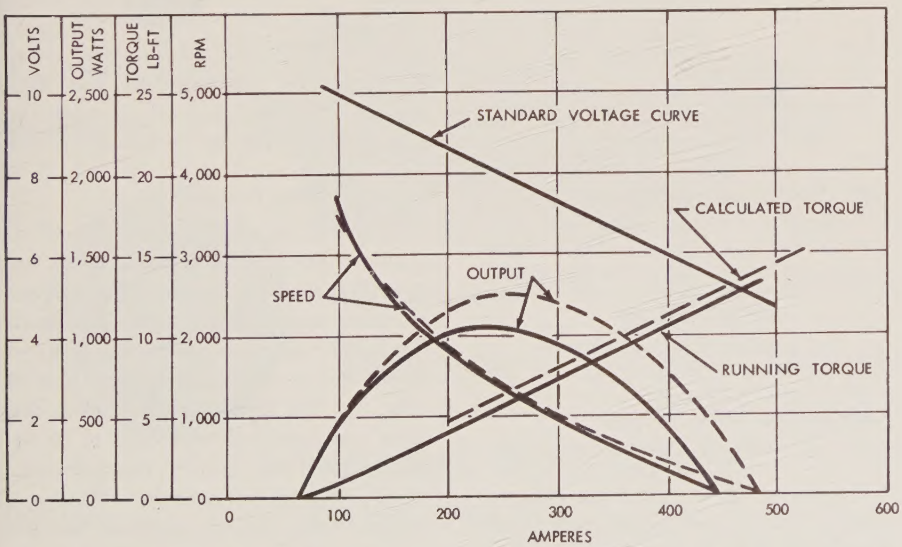


Fig. 6—This graph shows the performance characteristics of the 12-volt cranking motor at 0°F. The graph must be corrected for other temperatures. The running torque is determined by subtracting 0.7 lb-ft from the calculated torque curve. The 0.7 is an empirical factor based on experience with similar motors. The dashed lines show greater speed and higher output obtained with lower resistance fields. The standard voltage curve is for a 12-volt battery and takes into consideration an allowance for voltage drop in the cables connecting the battery and cranking motor.

versely to any great extent. At high currents the series fields were still more than enough to saturate the iron. The major effect was from the current draw of the shunt field—approximately 8 amp.

An interesting sidelight concerned the free-speed current. Since the free speed of a compound motor is less, the natural conclusion is that the current will be less. Actual results indicated, however, that with all of the shunt excitation on one pole, the flux density in the teeth under this pole at free speed was great enough to cause high iron losses and, consequently, high current to supply the energy for these losses. This, of course, did not detract from the useful output.

Another interesting difference proved to be beneficial. The overall resistance of the motor was reduced, since one of the series fields was eliminated. Accordingly, with reduced iron losses and more armature current draw, the peak output of the motor was increased, in one comparison, from 920 watts to 1,050 watts. This was enough to crank any passenger car engine in 1954. When some engines were increased to a 370 cu in. range of displacement and 10.5 compression ratio, the cranking motor was no longer adequate, especially under hot-stall conditions of the engine. However, the cranking motor did not have to be increased in overall size to give the required performance. In the original design, endless combinations had been tried to give the best proportions and most effective use of material. For minimum mean length of turn, the series field coils had been given maximum radial depth and minimum circumferential spread, leaving air space between coils. When this space was filled by making coils of thicker copper strap, the motor speed was greatly increased and peak output raised to 1,250 watts (Fig. 6).

Speed Determination

The calculations for cranking motor speed were based on 12-volt battery characteristics. The applied voltage on a cranking motor decreases markedly with an increase of current draw because the battery is a very limited power source with an internal resistance which is very high in comparison with other usual sources for this amount of current. A standardized 12-volt battery curve (Fig. 6) is always used for calculations to compare cranking motor designs. The curve

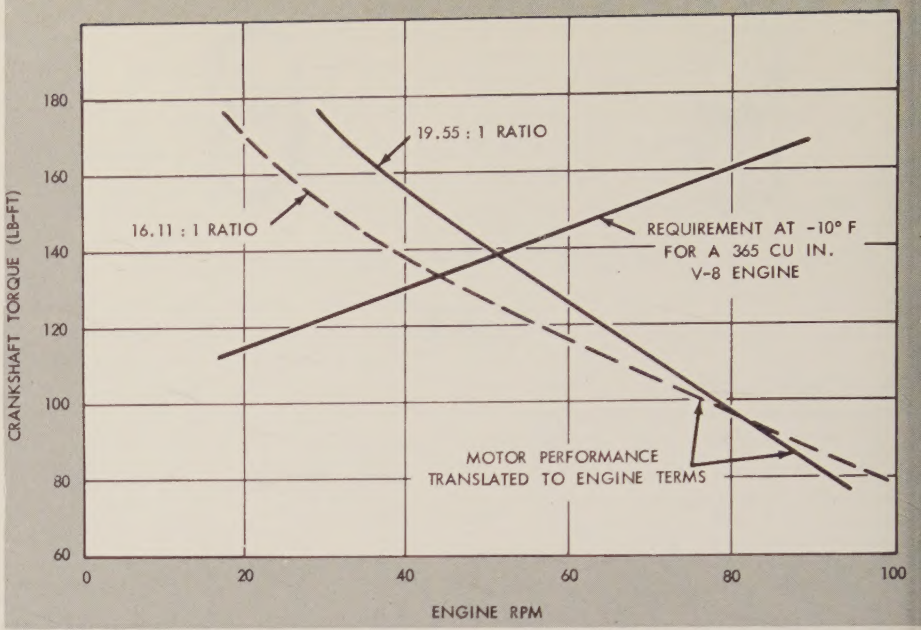


Fig. 7—Plotted on this graph is a curve of engine speed versus required starting torque for a 365 cu in. V-8 engine starting at a temperature of −10°F. Also plotted are adjusted values of torque versus speed for 2 different cranking ratios. Intersection of the cranking motor performance curve with the engine requirement curve gave the cranking speed. Also indicated was the effect of different cranking ratios. The high-speed, 12-volt cranking motor requires more gear reduction for effective use. A special 12-14 pitch pinion gave an increase in cranking speed from 45 rpm to 52 rpm on a typical 365 cu in. engine at −10°F. As can be seen, the gain in cranking speed is greater as cranking becomes more difficult. Since motor torque is less, the battery current drain is also less.

includes an allowance for the drop in the cables connecting the battery and cranking motor. The points on the speed curves (Fig. 6) were calculated by the following formulae:

$$E_b = E_t - IR_a - IR_f - E_{bd} \qquad (1)$$

$$N = \frac{(E_b)\ (p)\ (60 \times 10^8)}{P\ \phi\ Z} \qquad (2)$$

where

E_b = back emf (v)
E_t = motor terminal voltage obtained from standardized 12-volt battery curve (v)
I = motor current (amp)
R_a = armature resistance (ohm)
R_f = field resistance (ohm)
E_{bd} = brush drop obtained from curves of brush drop versus current density, for grade of brush used (v)
N = motor speed (rpm)
p = number of current paths in armature
P = number of poles
ϕ = lines of flux per pole
Z = number of armature conductors.

Correcting motor speed for different battery, circuit, and temperature conditions is readily done by ratio of back electromotive forces. In equation (1), however, several items must be adjusted. The motor terminal voltage E_t depends on the battery size, the size and length of connecting cables, and switch contact drops. Thus:

$$E_t = E_{bat} - IR_c - E_{cont} \qquad (3)$$

where

E_{bat} = battery terminal voltage at the particular temperature and current draw (v)
I = motor current (amp)
R_c = resistance of cables (ohm)
E_{cont} = switch contact drop (v).

The armature resistance R_a and field resistance R_f also must be corrected for ambient temperature.

Engine Cranking Performance

A specific program was outlined to evaluate the cranking performance of the 12-volt cranking motor. The engine cranking requirements were determined from cold room tests, and curves of speed versus the cranking torque were plotted (Fig. 7). The cranking motor performance was corrected according to its resistance and the battery characteristics at the chosen temperature. Values of torque versus speed, adjusted for the cranking ratio, also were plotted on the same curve. Intersection of the motor performance curve gave the cranking speed. This same method also told the effect of different cranking ratios.

Mechanical Design Completed the Overall Task

There were many new mechanical design features incorporated into the 12-volt motor design. Some of them were basic to the proper functioning of the motor. To assure engagement of the fast accelerating motor and to reduce wear and damage to the ring gear and pinion, the clutch sleeve and the shaft have a reverse spiral spline (Fig. 8). The screw-jack effect of the spline forces the pinion into full mesh before cranking torque is applied. A special, austempered retaining collar over a snap ring on the shaft restrains this force at the end of the pinion travel.

For the higher cranking ratio desired (Fig. 7) a smaller, non-standard pinion was developed, since the size of the ring gear could not be increased. As 9 teeth seemed to be the lowest practical number for good tooth contact ratio, the smaller gear used finer pitch teeth, a modified 12-14 pitch instead of the former 10-12 pitch. With the pitch diameter changed from 0.90 to 0.75, the ratio increase possible (from $0.90/0.75 = 1.2$) was 20 per cent. Actual practical change was from 16.11 to 1 to 19.55 to 1.

The smaller teeth might be deemed frail. However, they have given better service than previously experienced. As a basic part of the design, austempering heat treat was specified for gear toughness. The helical spline relieves most of the shock loading.

To reduce the use of steel in the field frame and to have a smooth, straight bore, the frame was designed to be extruded with a thin section at the commutator end. The brush holders were mounted on the inside diameter of this thin section rather than in the conventional manner on the end frame for 2 reasons: (a) piercing openings to make connections to the field would entail extra expense and (b) without inspection openings, the connections have to be made and leads arranged in place before the bearing cover is assembled.

Many brush holder designs were proposed and discarded before a final design was selected (Fig. 8). The brushes are attached by screws to swivel arms swung in pairs from a support bracket. One of each pair of swivel arms is molded of plastic to support the insulated brushes. The spring is of cantilever design which acts mutually on the 2 arms of a pair. This design was justified, in spite of the high spring rate, by test results which showed that brush life and motor torque were not affected by the wide range of brush pressure—8 psi to 20 psi. Rapid brush wear resulted when the pressure fell below 8 psi at the finish of a test.

The connection from the fields to the solenoid through the frame is accomplished by a connector insulated by an elastic grommet, removing the complexity of washers and bushings required to insulate a terminal stud. In alternate designs, the field coil strap is brought through the grommet to connect to the solenoid.

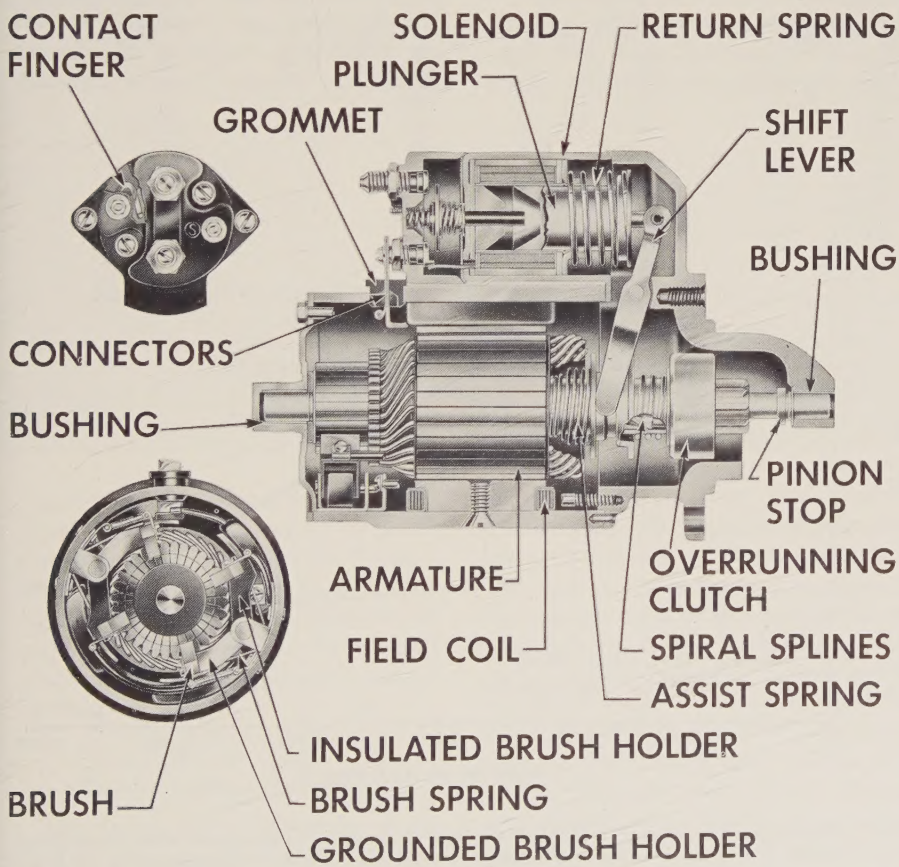


Fig. 8—This cross-section of the 12-volt cranking motor with an enclosed shift lever shows the splash proofing protection and the connection from the field coils to the solenoid through an elastic grommet. The assist spring function is also indicated. The end view illustrates the swivel brush holders and the cantilever spring common to a pair.

New Design Guards against Internal Condensation and Freezing

After the production plant had digested the basic tooling program essential to the change from a 6-volt to a 12-volt cranking motor, more design features were incorporated to add to customer satisfaction. The longest felt need had been for splash proofing to protect against corrosion of the solenoid switch contacts and ice formation on the contacts and shift mechanism. These problems resulted from the low mounting position of the cranking motor on a V-8 engine.

The use of the reverse spiral spline had improved the engagement characteristics to the point where less force was required on the clutch jump-spring. This meant, in turn, that the solenoid would not have to be so powerful. The idea was then proposed that perhaps the solenoid could be made small enough to pay for splash proofing. This idea was thought worthy of further consideration.

The first step was to establish a solenoid requirement curve (Fig. 9). The requirement was decreased by the conception of an "assist" spring (Fig. 8). The maximum force of the assist spring is substantially less than the minimum effect of the shift-lever return spring and does not interfere with proper and complete disengagement. When the solenoid acts to shift the motor into engagement, the force of the assist spring helps to overcome shift-lever return-spring and jump-spring forces.

Numerous electrical designs were made

to establish the most economical proportions to satisfy the solenoid requirements, keeping in mind that it was also desirable to reduce the solenoid current. The end result was a construction which gave the exact performance desired with smaller diameter and length of case, smaller diameter plunger, and smaller electrical coils.

With the electrical performance established, the solenoid was blended in with new mechanical design features of the motor to give the splash proofing desired and to overcome previous mechanical problems. Some of the design features are as follows:

- The solenoid case is made of drawn, seamless steel
- The flange of the solenoid case is attached directly to a radially extended "hood" of the drive housing which encloses the shift-lever mechanism. The throat of the hood is designed to avoid coring difficulties in the casting process
- A new type of gasket is used to assure its staying between the clamping surfaces of the solenoid cover and case during assembly
- The re-designed motor terminal of the solenoid has an internal thread which allows the field coil strap to be clamped directly against its end with a No. 10-32 screw
- The plunger has an internal taper, and its seat has an external taper to give maximum bearing length
- The lever-return spring is a compression spring.

The design of the cranking motor does not give complete sealing against water. However, the design is such that it would be difficult for water in any quantity to gain access to the shift lever and freeze. If the temperature were low enough for freezing, any entering water would freeze at possible entrance points, such as the clamped flange surfaces, and form a barrier against further flow.

In addition to the original problem regarding splash-proofing, there also was the problem of internal condensation. Moisture vapor present in the switch compartment would condense and freeze on the contact surface of the battery terminal of the solenoid. This occurred after the engine was stopped and the cold battery cable rapidly drew heat away from the terminal. The present design excludes water from inside the cover and the nearby coil case and keeps the vapor level inside the cover low enough to prevent formation of ice. A completely air-tight seal of the moving plunger would have involved friction losses in the performance of the solenoid.

Reverse-Spiral Spline Eliminates Pinion Adjustment and Allows Smaller Size Clutch

It was pointed out earlier that a reverse-spiral spline on the clutch sleeve and shaft was necessary to secure proper pinion engagement. The use of the spiral spline, in turn, was responsible for 2 other major developments.

One development concerned elimination of pinion adjustment which saved an assembly operation and the need for extra mechanical parts. Elimination of the pinion adjustment was based upon 3 factors: (a) more variation of pinion-to-flywheel clearance could be tolerated, since engagement was not as sensitive to jump-spring pressure, (b) the attachment of the solenoid directly to the drive housing removed some of the causes of variation, and (c) a statistical study of actual shop tolerances on the parts and a probability combination of these tolerances assured that only a very few would ever be outside the acceptable range.

The second development concerned the use of a smaller size overrunning clutch (Fig. 10). The smaller size was desirable for low inertia, material savings, and for clearance reasons on automatic transmissions used in 1956 and 1957 model cars. Factors which contributed to the use of a smaller diameter clutch were: (a) lower shock loading during engagement, (b) development of "accordion"-type springs in place of helical springs to hold the rollers in place, and (c) toughness resulting from austempering. The clutch sleeve was made shorter because the lighter jump-spring required less room.

Delco-Remy process development engineers contributed another improvement to the clutch design. One of the major assignments of process engineers is to devise methods of shaping metal by cold forming rather than by metal removal, such as turning and boring. In the case of the cranking motor, the clutch sleeve was changed from a drilled and broached screw machine part to a welded tube having swaged splines (Fig. 10). Through the cooperative efforts of product and process engineers, the internal spline was changed from a standard

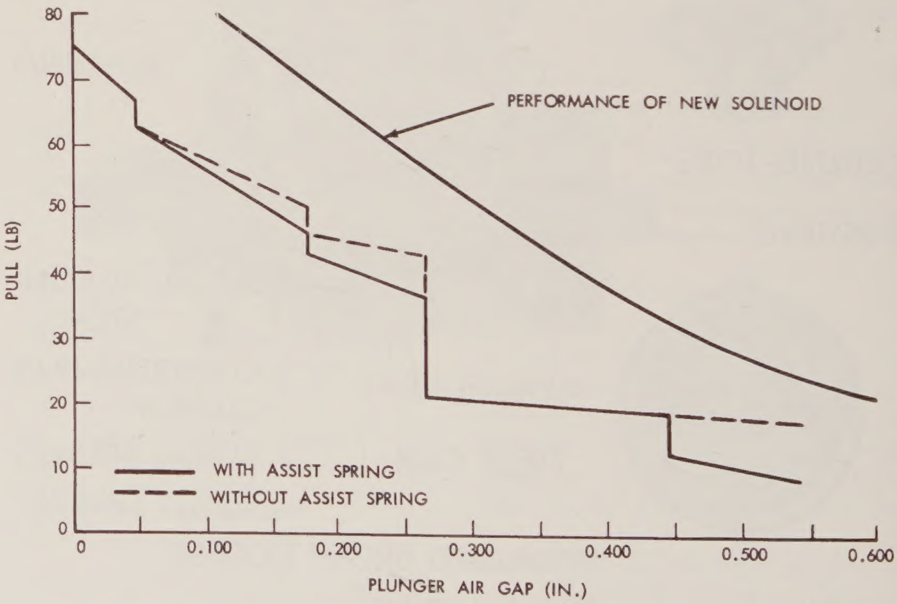


Fig. 9—The solenoid requirement is based on the total of the spring forces against which the solenoid must operate. The forces, in order as the plunger moves, are the shift-lever return-spring force, the clutch jump-spring, the contact assembly return-spring force, and the contact over-travel spring force. A novel feature of the 12-volt cranking motor design is the use of an assist spring to help the solenoid in the first part of the plunger travel, when the air gap is wide. The graph indicates the lesser requirements of the solenoid as a result of the use of the assist spring. The performance includes a margin for friction of the moving parts and heating of the solenoid coil.

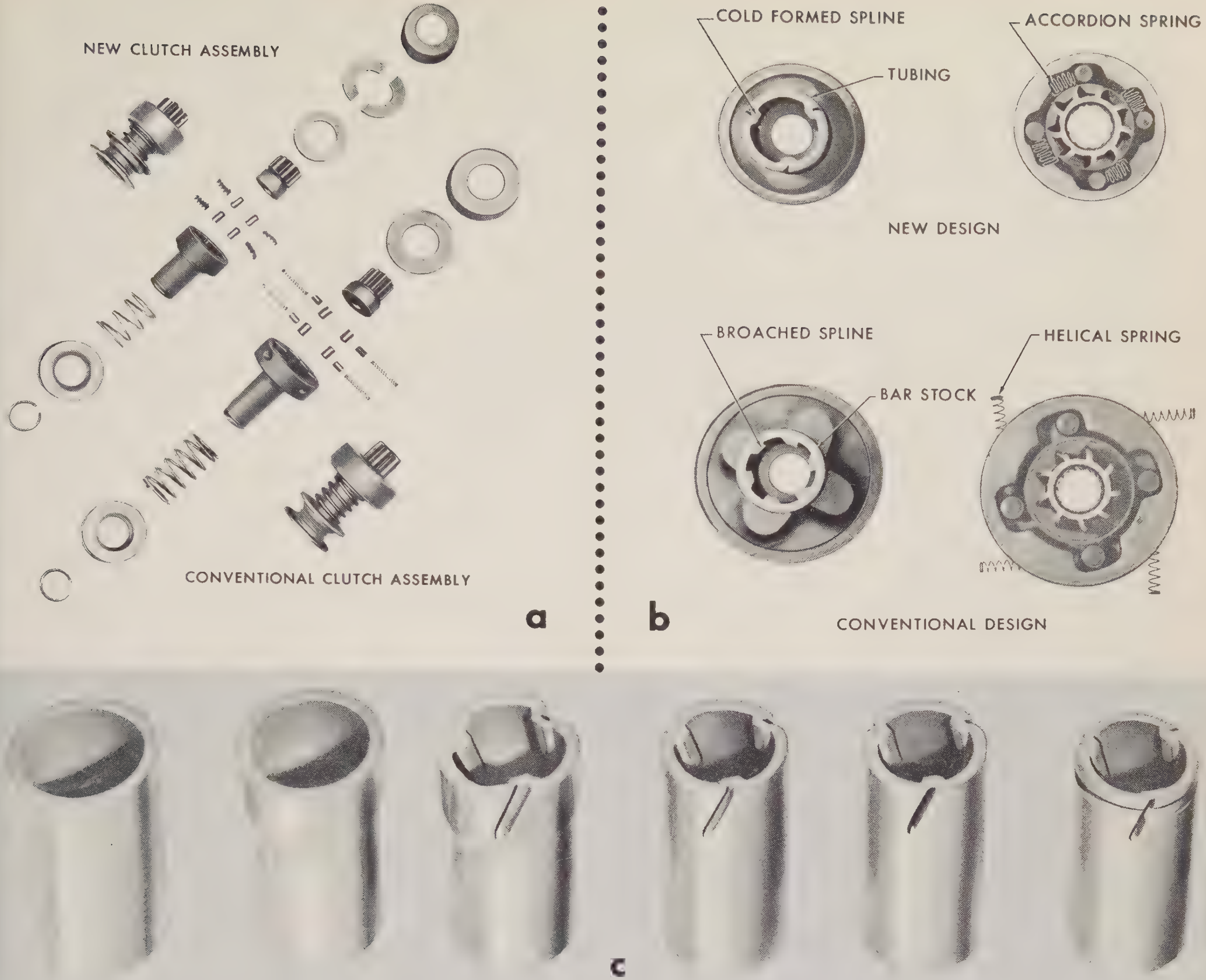


Fig. 10—The overrunning clutch component of the 12-volt cranking motor was re-designed to be of small diameter. View (a) shows the difference between the present and formerly used clutch assembly. View (b) shows the comparison between the old clutch and the smaller diameter clutch having an accordion spring. The clutch sleeve component of the clutch assembly was changed from a drilled and broached screw machine part to a welded tube having swaged splines. View (c) shows the sequence of cold forming operations to make the spline.

6-tooth form to a 3-tooth form. Calculations, tests, and comparison to pinion tooth strength showed that the 3 splines had the required strength. No such deviation had previously been considered with standard tools because the conventional machining methods produced 6 splines as readily as 3.

Cooperative Effort Resulted in Many Advantages

The design and development of the 12-volt cranking motor has incorporated an imposing list of production and tooling suggestions, all of which have contributed to the fulfillment of engineering and management objectives. These suggestions modified or added the following features: hot rolled laminations, extruded frame, drawn steel commutator end bearing support, round conductors, headed-bar commutator, plastic dip-coating of field coils and flexible leads, double support for brush holder pin, molded rod for solenoid contact assembly, elimination of pinion adjustment, smaller diameter and simpler clutch shell, cold-formed clutch spline, improved method of holding solenoid cover gasket in position, improved design of drive housing to facilitate casting and machining, and an overall design which facilitates assembly on conveyorized jigs rather than on a belt.

Numerous other instances of refinement could be included. Manufacturing objectives also helped account for retaining previous engineering features such as the phosphate coated shaft, welded connections in the fields, and high-copper brushes.

Summary

The end result of the 12-volt cranking motor developmental program has been an economical motor to crank the larger size automobile engines faster than ever before. Whereas rain and freezing weather had sometimes made the cranking motor inoperative, the 12-volt cranking motor is designed to assure successful operation under such adverse conditions.

Methods Engineers at Work:

Development of Semi-Automatic Assembly Equipment for Oldsmobile Front Suspension Control Arms

By ROBERT W. TRUXELL,
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Oldsmobile
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When the 1957 Oldsmobile ball-joint front suspension was released for production, the Methods Engineering Department undertook the development of a semi-automatic assembly machine to assemble shafts and bushings in the lower control arm. To approach the problem in a manner which would result in an economical and efficient method of assembly, the many interrelated physical and manual requirements were summarized in *plans of operation*. These plans were the composite results of experimentation and methods engineering analysis. The assembly machines which were designed, built, and installed incorporated all specifications of the plans of operation. The manpower utilization realized from the assembly machine project proved the soundness of applying basic methods engineering principles, through the use of plans of operation, to machine design development.

Plans of operation establish machine requirements

THE 1957 Oldsmobile ball-joint front suspension (Fig. 1) requires the assembly of shafts and bushings into the lower control arm. The bushing has a pilot section which guides it into the hole in the control arm. The pilot section serves to locate accurately the bushing concentrically with the hole during the threading operation. The bushing has external threads which cut mating threads in the control arm during assembly. A design requirement specifies that the bushing must be driven to a solid seat against a shoulder provided in the control arm.

As part of a continuous manufacturing improvement program, a study was made of various methods which could be used to perform this assembly operation on an automatic or semi-automatic basis. The production volume needed and the effort obviously required to drive the bushing into location dictated the use of power equipment. Further study indicated that a semi-automatic assembly machine was feasible, and the project was assigned to the Methods Engineering Department for development. The final design requirements for the assembly machine developed (Fig. 2) were based on specifications resulting from methods engineering analyses.

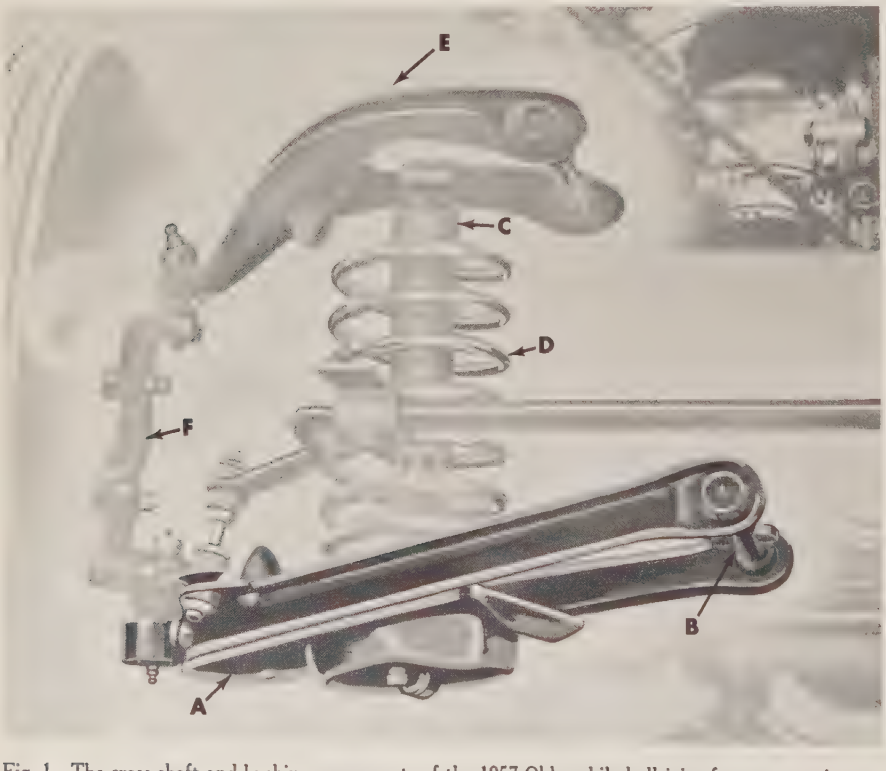


Fig. 1—The cross shaft and bushing components of the 1957 Oldsmobile ball-joint front suspension are assembled into the lower control arm by means of a semi-automatic assembly machine designed especially for the operation. The front suspension is composed of the lower control arm (*A*), lower control arm cross shaft (*B*), shock absorber (*C*), coil spring (*D*), upper control arm (*E*), and the steering knuckle (*F*).

Methods Engineers First Developed Plan of Operation

The first step in the development of the semi-automatic assembly equipment was to establish a *general plan of operation* for the complete work place area (Fig. 3). This plan established the relationship between machines, as well as the relationship between the man, the machines, and the flow of material. Estimates were made of the number of complete arm and shaft assemblies which could be produced by the semi-automatic assembly machines in a regular production situation. These estimates were the basis for determining the most economical installation, considering both standard time and initial investment.

The next step was to prepare *work place area* and *machine plans of operation* from the general plan of operation. The machine plans of operation (Fig. 4) specifically described the physical relationships re-

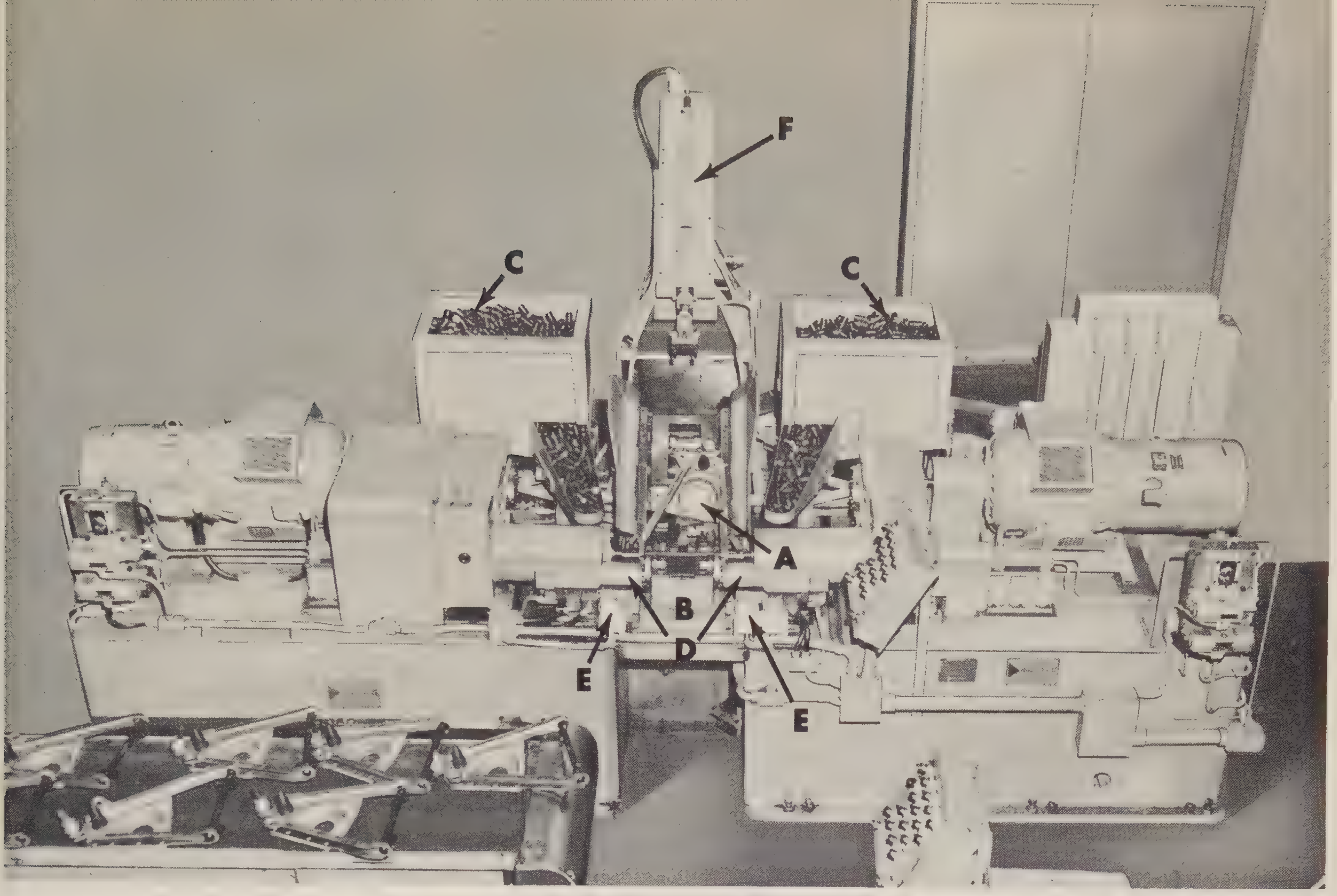


Fig. 2—This semi-automatic assembly machine was developed by methods engineers of Oldsmobile Division to assemble cross shafts and bushings into the lower control arm of the ball-joint front suspension. The control arm (*A*) is loaded first into the clamping fixture (*B*), and the bushings are loaded from hoppers (*C*) into the driving sockets (*D*). The operator then presses the cycle start buttons (*E*). This puts the machine in automatic cycle to drive the bushings. The assembly is disposed of automatically when the unload cylinder (*F*) retracts.

quired for the most efficient manual operations and the design requirements for each piece of assembly equipment.

Experimental Tests Determined What the Machine Had to Do

With the general plan of operation as a guide, a mock-up of the proposed work place area was then made to determine the best possible motion pattern and work place layout. The mock-up made it possible to estimate accurately the time elements of the operator's work cycle and, also, the location of controls which were to be mounted on the assembly machines. After the desired manual operations of the work cycle were established, the physical requirements of the machines were then determined through a series of experiments.

Physical Requirements

The physical requirements of the assembly machines were dictated by the following factors: (a) the bushing had to be driven until seated against the control arm, (b) an accurate relationship between the cross shaft and the control arm had to be maintained for optimum caster and camber adjustment, and (c) the control arms had to be pre-stressed during assembly to maintain the torsional requirement of free rotation of the cross shaft after assembly (Fig. 5).

Torque and Bushing Seating Requirements

The torque necessary to drive the bushing until it was seated against the control arm was established by first assembling the bushings in the control arms with a hand-operated torque wrench. Torque versus revolution curves were then plotted (Fig. 6). The curves indicated that the maximum torque required to drive the bushing occurred approximately 2 revolutions before the bushing was seated on the control arm shoulder. It also was found that an objectionable burr would be created between the shoulder and bushing if the bushing were seated against the shoulder with the same torque as that required to drive the bushing. These data led to the conclusion that if the bushing were to seat against the shoulder in such a way as to produce an acceptable assembly, the torque at the time of bushing seating would have to be reduced to approximately 100 ft-lb during the last revolution of the bushing. It was felt that such a seating torque would produce a burr that would permit visual inspection of the bushing seating condition and still not create a burr large enough to be considered a safety hazard.

Pre-Stressing Requirements

A testing machine was used to obtain information for plotting curves showing force versus deflection for spreading of the control arm (Fig. 7). This information was necessary to establish the pre-stressing force required to prevent the shaft from binding in the bushings after assembly. With a dial indicator set to show deflection and a small jack screw to apply a spreading force, control arms were hand assembled first and then tested to determine the torsional force required to rotate the cross shaft. It was found that by spreading the control arm with a force of approximately 20 lb the resulting assemblies would have the desired freedom of cross-shaft rotation.

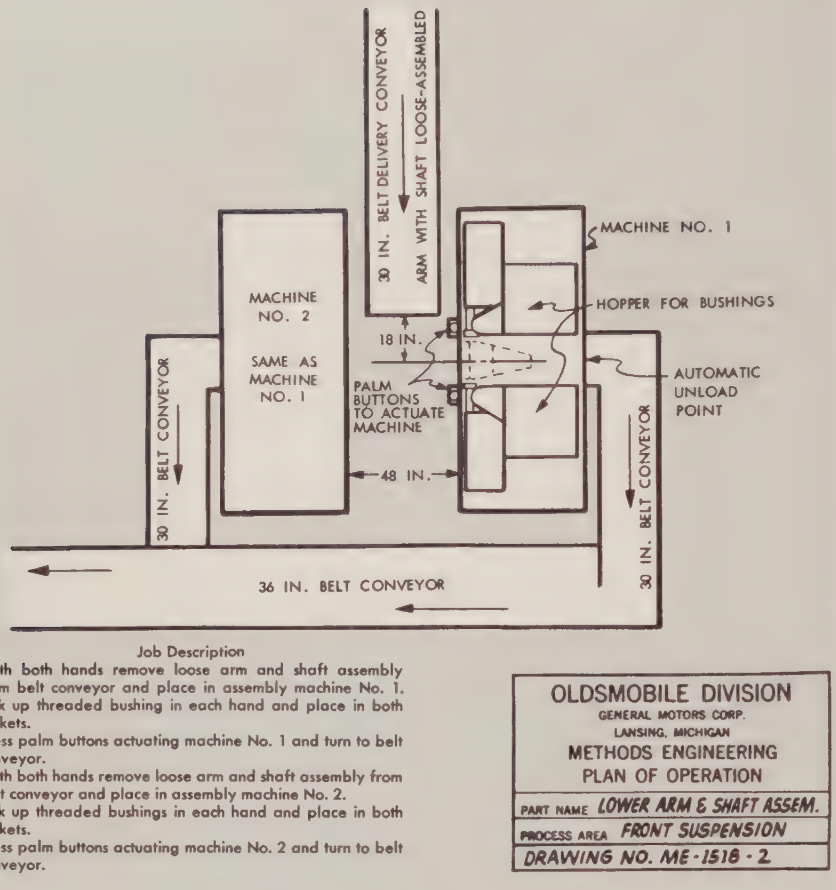


Fig. 3—This is the *general plan of operation*, prepared early in the development of the semi-automatic assembly machine to describe the basic physical relationships of the work place area. Each operation was allotted an estimated time value. Thus, a value for possible production per hour was determined.

The location point on the control arms at which the pre-stressing force is applied has a great deal of influence on the tightness of the cross shaft. It was determined that for the most consistent results the pre-stressing force should be applied on the inside face at a point between the hole and the outside edge of the control arm. It was required that the pre-stressing force remain constant during the complete assembly operation.

Design of Assembly Machine Based on Plan of Operation

After completion of the basic plans of operation, the assembly machine design requirements were given to an outside supplier. All basic design work then was done in accordance with the machine specifications shown on the machine plan of operation.

Torque Adjustment and Driving Force

With a given torque range of 75 ft-lb to 400 ft-lb, the proper type of driving power for the assembly machine became of major importance. The use of air-powered tools for driving nuts, screws, and bolts was considered. Past experiences on similar operations, however, indicated that power losses through wear, slow driving speeds, and high maintenance costs were likely. A more positive and dependable driving force, therefore, was required for the assembly machine.

A survey of the types of drives which were adaptable to this assembly operation resulted in the selection of a 10 hp, standard squirrel-cage motor having an eddy-current clutch and eddy-current brake complete in one housing. Torque and speed are controlled by changing the excitation of the eddy-current clutch. The eddy-current brake serves to increase the rate of deceleration, as well as provide a constant low torque for seating the bushing.

The excitation of the eddy-current clutch is automatically governed by a small generator on the output shaft of the unit. Any increase in output shaft speed due to a reduced load increases the generated voltage which, in turn, decreases the excitation of the eddy-current clutch, thereby reducing the output speed. By limiting the maximum excitation applied to the clutch, the maximum output torque of the unit also is controlled. When this torque limit is exceeded by the external load, the unit stalls. This stalling characteristic is desirable since it permits setting the unit to stall at a pre-determined maximum torque. This prevents machine damage due to cross threading of the bushing or other stock defects.

The maximum torque output of the unit can be varied during any part of the machine cycle by an electronic control circuit. The unit, therefore, is ideally suited for the bushing assembly operation because of the 2 controlled torques required: (a) a high torque to drive the bushing and (b) a low torque for seating the bushing.

The speed and torque control characteristics plus the low maintenance requirements of an electric motor and coupling made the overall unit ideal for driving and seating the bushings in the control arm.

Driving Socket Movement

The assembly machine incorporates a driving head having individual units consisting of the driving motor, eddy-current couplings, gear reducer, and socket spindle, which are readily accessible for maintenance.

During the assembly operation, the work piece is stationary. The unit for driving the bushing travels horizontally to the work piece. The horizontal travel of the driving spindle is accomplished by a sliding spline which is moved in and out through a gear reducer by a hydraulic cylinder. Speed control of the horizontal travel is obtained through flow control valves placed in the exhaust lines of the hydraulic cylinder.

Driving Speed

Based on the process cycle time established in the plan of operation, a driving

Specifications

1. Each nut runner to have a variable torque range from 75 ft-lb to 400 ft-lb.
2. Each nut runner to have a variable speed range from 10 rpm to 160 rpm.
3. Each nut runner must have automatic controls to change high driving speed and torque to low seating speed and torque.
4. Positioning mechanism (hydraulic cylinder) shall be of sufficient capacity to apply 40 lb of force to each arm for pre-stressing during assembly. Force must be variable from 5 lb to 40 lb to insure correct shaft rotational tightness.
5. Magnetic sockets are required to keep the threaded bushings in line with the inner shaft when starting assembly.
6. Palm buttons shall be inter-connected and must both be depressed until pinch point has been guarded.
7. The complete machine cycle, including unload time, must meet a selected time value based on production requirements.
8. The machine shall be equipped with an automatic unloader which discharges the completed assembly out the back of the machine onto a belt conveyor.
9. Lockouts must be provided on the unloader circuit to permit past model service parts to be assembled on this machine.

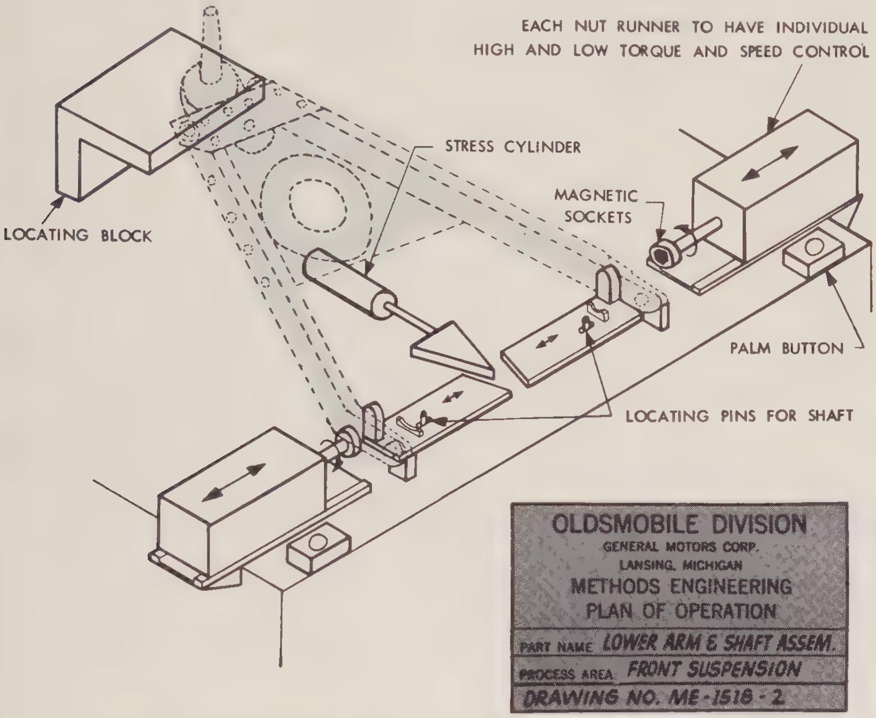


Fig. 4—Shown here is the *machine plan of operation* prepared after the general plan of operation was finalized. The machine plan describes the general machine requirements for product quality and efficient manpower utilization.

speed of approximately 135 rpm is used. This speed is low enough to prevent excessive heat from being generated while the bushing is threading itself into the control arm. The gear reducer developed for the assembly machine has a 7.2 to 1 reduction ratio which provides a maximum driving speed of 155 rpm with 100 per cent excitation of the eddy-current coupling. The driving unit has a maximum full load output speed of 1,120 rpm due to an approximate 7 per cent slip of the eddy-current clutch and brake.

Sequence of Machine Operations

To adhere to the machine plan of operation, proper clamping and prestressing of the work piece is required before the actual assembly takes place. These requirements are accomplished by the following sequence of events:

(1) When all parts to be assembled are properly located in the assembly fixture, the operator depresses and holds 2 start-cycle buttons

(2) When the start-cycle buttons are depressed, electrically operated solenoid valves are energized to direct hydraulic oil to a shaft clamp cylinder and an elevator cylinder. A limit switch at the end of the elevator cylinder stroke energizes a timer. This timer permits a static weight, which is prestressing the control arm (Fig. 8), to come to equilibrium before any additional cylinder sequences take place. After this time delay, solenoids are energized to direct oil to the stress-slide locking cylinder and to both driving-head feed cylinders. As the driving heads advance under fast feed, a limit switch is energized which, in turn, excites the eddy-current coupling. This starts the output shaft in high rpm. The limit switch also energizes the control valve solenoids which direct the oil through flow control valves, placing the driving heads in slow advance

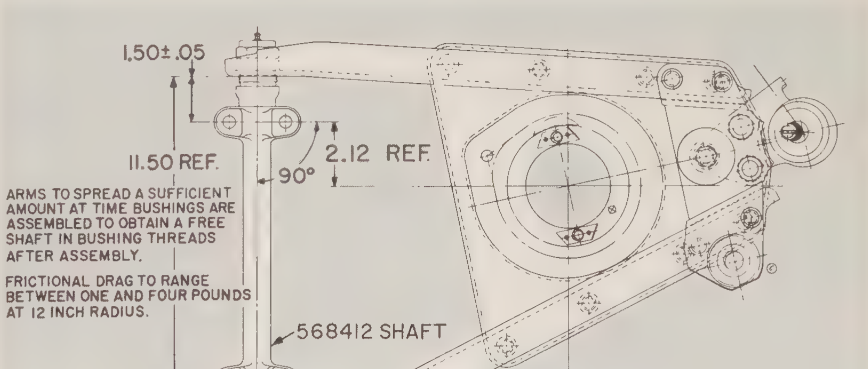
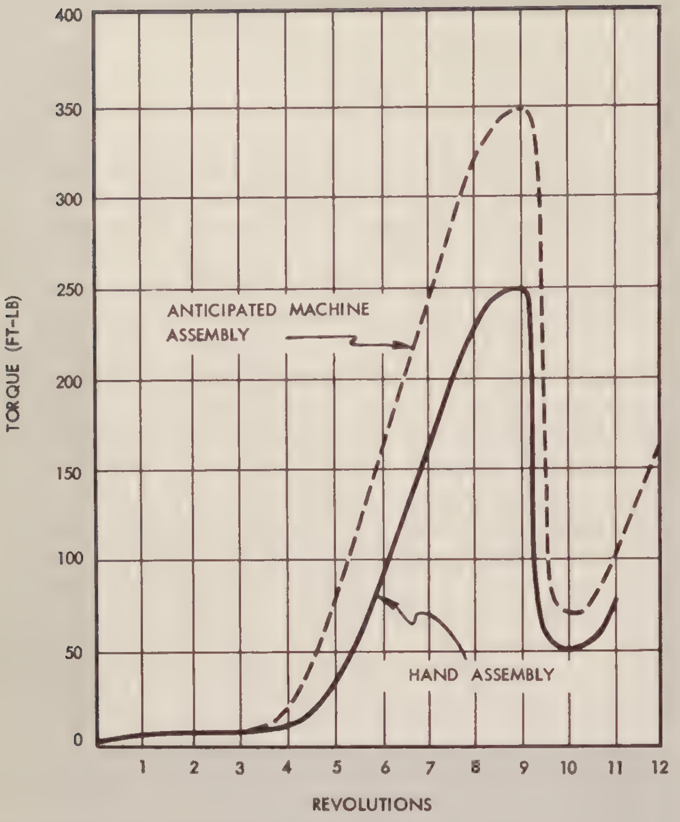


Fig. 5—This design drawing illustrates the assembly of the bushing, cross shaft, and lower control arm used on the front suspension of the 1957 Oldsmobile. The control arm oscillates about the cross shaft with vertical movement of the front wheels and is attached to the control arm by means of 2 bolts.



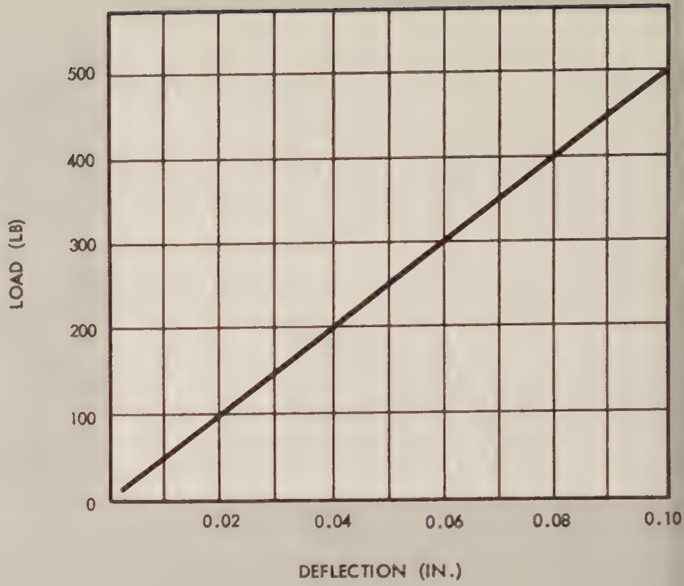


Fig. 7—With the aid of a testing machine, a shaft deflection versus force curve was plotted to aid in establishing the pre-stressing force required to prevent the cross shaft from binding in the bushings after assembly. This force is necessary to compensate for "spring-back" of the control arm after assembly so as to allow the cross shaft to oscillate freely in the bushings.

Fig. 6—The torque and revolutions required to drive a bushing until seated against the control arm were determined by using a hand-operated torque wrench to assemble the maximum allowable outside diameter bushing to a control arm having the minimum hole diameter. The resulting curve (solid line) indicated that for approximately the first 4 revolutions the torque required was negligible while the bushing was threaded onto the cross shaft but increased rapidly as the bushings cut threads into the control arm. The torque dropped off as the thread cutting was completed and then increased as the bushing was seating against the control arm. The driving torque output anticipated from the semi-automatic machine is indicated by the dashed line.

(3) As the driving heads advance, each driving spindle feeds into the work under the preset upper torque limit until a limit switch is contacted one revolution before bushing seating (Fig. 9). As this action takes place, the unload cylinder moves forward into the unload position. At this point, forward travel of the driving spindle is stopped, the electrical excitation to the eddy-current coupling is reduced to the preset bushing seating torque, and an electrical timer is energized. The driving spindle continues to drive the bushing through the additional revolution and then stalls

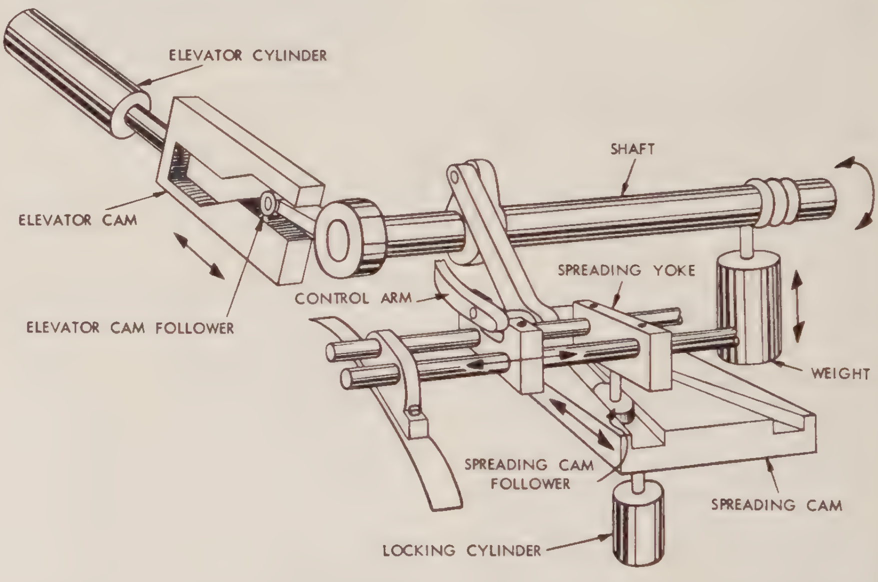


Fig. 8—The basic design of the mechanism for pre-stressing the control arms is illustrated in this schematic drawing. A constant, uniform pre-stressing force is attained by utilizing a suspended weight as the stress force. (Inaccuracies due to friction losses are negligible.) To start the pre-stressing operation, with a control arm properly located in the fixture, the elevator cylinder is actuated to move the elevator cam forward. With the elevator cam forward, the elevator cam follower moves first to the lower and then the free position, after the spreading force applied to the control arm equals the weight. When viewed from the cam follower end of the shaft, counterclockwise rotation of the shaft activates the spreading cam which, in turn, moves the spreading yokes outward. The only variables in the linkage that could affect uniformity of stress are the spreading cam and follower contact friction. After the spreading cam plate has completed its movement, the locking cylinder is actuated to lock the spreading cam in its full-stress position. When the stressing action is reversed, the locking and elevator cylinders return to their normal positions, thereby raising the weight. The next cycle is then ready to start. This diagram shows only the left spreading-yoke assembly.

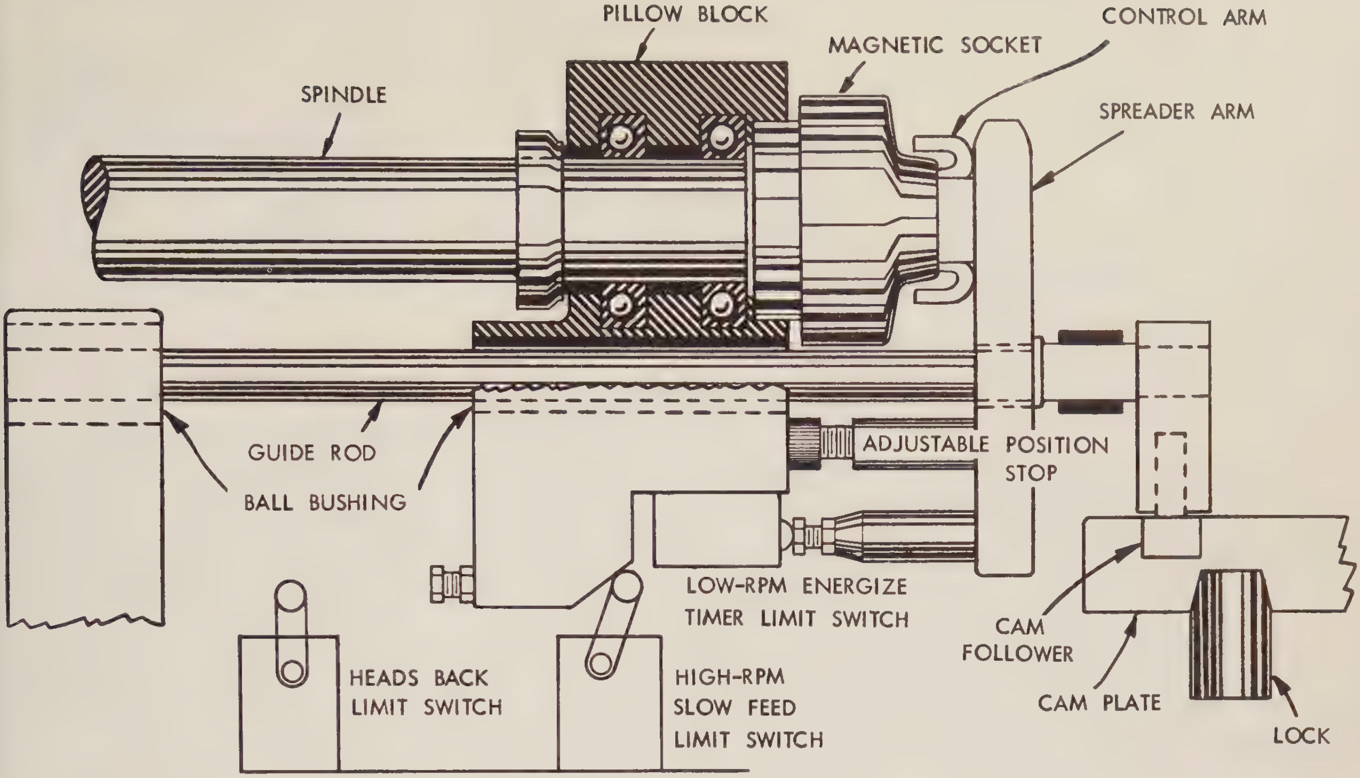


Fig. 9—During assembly of the bushing into the lower control arm, it is necessary to control the driving torque as the bushing is seated against the control arm. This is done by the use of a limit switch which is tripped one revolution before the bushing is to be seated against the control arm. The limit switch stops the high-speed forward travel of the driving spindle and reduces the excitation of an eddy-current brake and clutch (not shown) to the value required to provide a low torque for seating the bushing. The spindle drives the bushing through the remaining revolution until it is seated against the shoulder of the control arm.

as the bushing seats against the shoulder. The electrical timer, which is set for a time interval slightly longer than the time required to seat the bushing, then times out and de-energizes the locking cylinder, elevator cylinder, and driving head cylinder solenoid valves. All cylinders then retract to their original position. At this point in the cycle the automatic unload cylinder is returned to remove the assembled part from the fixture and place it on a belt conveyor at the back of the machine.

Driving Torque Adjustments

Before the assembly machines were put into operation, the maximum torque setting on the driving heads was set just above the maximum torque requirement. This maximum torque occurred when the largest allowable outside diameter bushing was assembled to a control arm having the smallest allowable hole diameter. Possible torque attainable was kept as low as practical to insure stalling and to prevent equipment damage whenever overloading occurred.

The minimum torque setting was made by starting with a torque lower than required and gradually increasing the torque setting until a satisfactory bushing seating condition was obtained. After completing the torque and speed adjustments, readings were recorded to provide a base starting point in case the originally adjusted machine characteristics were lost through normal maintenance, tube weakness, or failures.

Shaft Looseness

The automatic clamping and prestressing incorporated into the design of the assembly machine produced a uniformly stressed arm with a resulting uniformity of shaft freedom of rotation.

Bushing Seating

Seating of the bushing against the lower control arm has proven to be very successful. Since the bushing is seated against the arm with a controlled torque, the tolerance stack-up of the parts to be assembled has no objectionable effect on bushing seating quality.

Conclusion

The development of the semi-automatic assembly machine is a typical example of how the methods engineer works to improve quality and ease of manufacture while lowering cost. The project presented a problem which depended upon close cooperation between product engineering and manufacturing organizations for its solution and illustrates the steps necessary from the establishment of production goals and proposed method of operation to the determination of machine and operational specifications.

The semi-automatic assembly machine has served to produce a high quality product and has reduced the manual operations normally required for an assembly operation of this nature.

Actual production studies show that the average production rate equals 98 per cent of the estimated and planned goals. These objectives have been realized by first establishing a machine and work place plan of operation and then designing assembly equipment to meet those specifications.

A Mathematical Tool in Industry: An Algorithm for Curve Fitting by the Method of Least Squares

By LEO MARCUS
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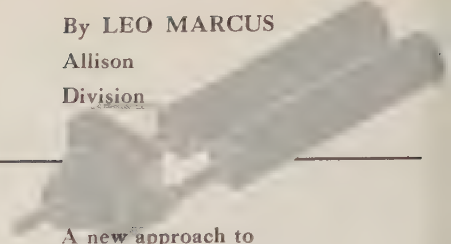
Division



A new approach to the classical theory of least squares

In its simplest form, experimental data is dependent upon two related variables X and Y which are plotted with the X values as the abscissa and the Y values as the ordinate. If the plotted points approximate a straight line, it becomes necessary to determine the coefficients of the analytical equation of the line which best fits the data. One approach to determining coefficients is the application of the method of least squares. In general, however, the dependent variable is dependent on more than one independent variable, and the form of the plot of the dependent variables versus any one of the independent variables is more complicated than a straight line. For the more general case, the *exact* form of the analytical equation to be fitted must be chosen beforehand before the classical method of least squares may be used. A lengthy iteration process must be performed in order to minimize the number of terms in the equation. To eliminate this shortcoming, a new approach to the problem of determining an equation to fit a set of experimental data was developed. The result has been a method, based upon the principles of the method of least squares, which (a) eliminates the trial and error computations required to determine the approximate minimum number of terms in the equation, (b) avoids the solution of an almost dependent set of simultaneous equations, and (c) has the advantage that any number of independent variables can be considered at the same time. This method has been successfully applied at Allison Division to such phases of experimental engineering as the design of fuel control systems, compressors, and turbines for turbo-prop and turbo-jet engines.

THE importance of determining an analytical function to fit a set of experimental data has long been recognized. Efforts have been made in the past to find reliable methods for relating an analytical function to a set of coordinates which, at the same time, avoid the dangerous loss of significant figures in the computation.

It is well known that the classical theory of least squares is one of the best methods for fitting an analytical function to a set of experimental data. This theory states that the best value of a quantity which can be obtained from a set of observed data will be the value for which the sum of the squares of the errors, or deviations, of the observed data from the best value is a minimum. The practical application of this theory is referred to as the method of least squares. This method is used to determine the arbitrary constants in an analytical function which is assumed to fit best a given set of observed data.

The method of least squares has one disadvantage—the exact equation to be fitted to a set of experimental data must be chosen beforehand. As a result, the final form of an equation using the minimum number of terms required can be obtained only by a method of trial and error.

New Method Eliminates Trial and Error Calculations

The one disadvantage of the method of least squares can be somewhat of a handicap when more than one independent variable is involved. This proved to be the case at Allison Division where experimental data involving two or more independent variables gathered from a wide variety of engineering developmental tests must be analyzed. This analysis usually requires the determination of an analytical function to fit the plotted points of the observed data. The amount of time spent in trial and error calculations necessary to determine the required analytical function expedited the development of a new method which would eliminate the disadvantage of the method of least squares. Even disregarding this one disadvantage, the size of the matrix which had to be inverted became unwieldy, making it impossible to solve for the set of coefficients.

The method developed for determining the approximate minimum number of terms in an analytical equation to fit a set of observed data eliminates the trial and error computations previously required for such a determination. Also, the new method has the advantage of avoiding the solution of a high order, almost singular set of simultaneous equations.

To explain fully this new method of curve fitting, the following mathematical derivation is presented. It will be assumed that the reader has a basic understanding of the principles involved in the method of least squares.

Mathematical Derivation

In the following derivation, the dependent variable Y is considered to be a single valued function. The restriction of this postulate may be removed by proper formulation of the analytical function to be fitted to a given set of data. Any number of independent variables X_i may be considered in formulating the possible analytical function to be fitted.

The form of the equation now can be formulated with the dependent variable Y on the left-hand side of the equation and all possible terms involving the independent variables X_i stated on the right-hand side as follows:

$$Y = f\,(X_1, X_2, X_3 \ldots . X_k). \qquad (1)$$

A simple means of stating the right-hand side of equation (1) is to fit first the dependent variable Y as a function of each independent variable X_i separately. This implies the following equation:

$$f\,(X_1, X_2, X_3 \ldots . X_k) = G_1\,(X_1)\,G_2\,(X_2)\,G_3\,(X_3) \ldots . G_k\,(X_k). \qquad (2)$$

It also is assumed that the function $G_i(X_i)$ shall be linear in its coefficients c_i. This gives the following equation:

$$G_j = c_0 + \sum_{i=1}^{m-1} c_i\,Z_i \qquad (3)$$

where

m = maximum number of terms

Z_i = a "one term" function of X_j. For example, $Z_1 = X_j$, $Z_2 = X_j^4$, or $Z_3 = \sin X_j$.

The final form of the dependent variable function Y now may be stated in the following linear form of the a_i terms:

$$Y = a_0 + \sum_{i=1}^{n} a_i V_i \qquad (4)$$

where

$$V_i = V_i (X_1, X_2, X_3 \ldots X_k).$$

It is now necessary to solve for the set of coefficients a_i ($0 \leqq i \leqq n$) such that the analytical function represents the "best fit," in the sense of least squares, and that the number of the $a_i \neq 0$ shall be minimized as much as possible. It is convenient at this point to translate the axis of the n dimensional space so that the origin of each axis occurs at its arithmetical mean. In this regard, equation (4) may be restated as follows:

$$Y - \overline{Y} = \sum_{1}^{n} a_i (V_i - \overline{V}_i) \qquad (5)$$

where the bar over Y and V_i denotes the arithmetical mean of the particular quantity computed from the observed data.

In comparing equations (4) and (5) it is seen that a_0, which is a measure of the true origin from the centroid of the observed data, is eliminated. The a_0 term may be computed as follows:

$$a_0 = \overline{Y} - \sum_{i=1}^{n} a_i \overline{V}_i. \qquad (6)$$

At this point in the derivation it should be stated that, for simplicity in notation, a parameter minus its arithmetical mean will be denoted by the lower case letter of that parameter—for example, $P - \overline{P} = p$.

From statistics, a measure of "goodness of fit" of a straight line is noted as a reduction in the sum of the squares of the ordinates due to regression, abbreviated *SSR*. If the straight line fits the observed data exactly the sum of the squares of the ordinates *SSY* minus the *SSR* is identically zero. This implies that the straight line which best fits a set of observed data is the one whose *SSR* is a maximum. An equation for a straight line based on this implication is as follows:

$$SSR = \frac{(\Sigma yv)^2}{\Sigma v^2}. \qquad (7)$$

Some authors use the following notations for the straight-line formula expressed by equation (7):

$$SSY = \sigma_y^2$$

$$SSR = \sigma_{ey}^2$$

$$SSY - SSR = S_y^2 = SSy.$$

With the establishment of the equation for a straight line, the reduction in the ordinates due to regression $(SSR)_i$ can now be computed for each v_i. It is assumed that the v_j which has the greatest effect in reducing the sum of the squares of the ordinates, or an error curve plot, is the predominate term in the analytical function which is being fitted to the observed data. The coefficient of regression which produces the largest $(SSR)_j$ is given by the following equation:

$$m_i = \frac{\Sigma y v_i}{\Sigma v_i^2}. \qquad (8)$$

It should be noted here that in the method of least squares the slope of a line passing through the origin also is given by equation (8).

The coefficient of the v_j which gave the largest *SSR* at this stage of the reduction can now be eliminated from equation (5) by the following formula:

$$y - \frac{(v_j)(\Sigma y v_j)}{\Sigma v_j^2} = \sum_{i=1}^{n} a_i \left[v_i - \frac{(v_j)(\Sigma y v_j)}{\Sigma v_j^2} \right] \qquad (9)$$

where

$$i \neq j.$$

The above process, starting with equations (7) through (9), is continued until criteria for the "goodness of fit" have been satisfied.

There are various methods used to measure the "goodness of fit" of an analytical function to a set of discrete data. Three such methods are as follows:

(a) Using the original data, the sum of the squares of the errors *SSE* shall be less than some predetermined value

(b) The absolute value of the maximum error at any of the original data points shall be less than some predetermined value

(c) The reduction in the *SSE* due to one additional term in the analytical function shall be less than a certain percentage of the *SSE* without considering this additional term.

In general, criterion (b) should be used if the given data is for a predetermined curve whereas criterion (c) should be used when given a set of random experimental data.

When all coefficients of terms used in the final analytical function have been eliminated from equation (4), by using equations (7) through (9), all the additional coefficients are made identically zero. The value of the coefficients $a_i \neq 0$ are obtained by the following equation in reverse to the order in which they were eliminated:

$$a_i = \frac{\Sigma y v_i}{\Sigma v_i^2} - \sum_j \left(a_j \frac{v_j v_i}{v_i^2}\right). \qquad (10)$$

It also can be shown that the value of the coefficients a_i obtained by this method would be identical to those obtained by the method of least squares, when consideration is given to the final form of equation (10).

The coefficient of correlation r_j, which is defined by the equation $r_j = \sqrt{(SSR)_j/(SSY)}$, is a measure of association between Y and another variable. The absolute value of the coefficient of correlation is less than or equal to 1. It should be noted in this derived method of fitting the analytical function, the choice of the term with the largest *SSR* is just a means of maximizing the coefficient of correlation between Y and one other term in the equation.

Numerical Examples

To demonstrate the derived method for finding a minimum number of terms in an analytical function to fit a set of observed data, 3 examples are presented.

Example I

The first numerical example is concerned with determining which single term on the right-hand side of the following equation will have a minimum variance:

$$Y - \overline{Y} = \sum_{i=1}^{7} a_i (X^i - \overline{X}^i). \qquad (A)$$

It is further required to compute the standard deviation and coefficient of correlation of the single term and also to express equation (A) in the form of the following equation for a specific set of data which have been plotted (Fig. 1):

$$Y = a_0 + a_j X^i. \qquad (B)$$

An equation in the form of (B) is useful when it is desirous to obtain the derivative of Y with respect to X subject to

the restriction that the first and second derivative shall be monotonic.

The first step is to express equation (A) in the form of equation (4) as follows:

$$Y = a_0 + \sum_{i=1}^{7} a_i V_i \qquad \text{(C)}$$

where

$$V_1 = X \qquad V_4 = X^4$$
$$V_2 = X^2 \qquad V_5 = X^5$$
$$V_3 = X^3 \qquad V_6 = X^6$$
$$V_7 = X^7.$$

The next step is to compute the arithmetical means of all the parameters (Table I-A). The original data is then adjusted for the arithmetical means such that equation (C) is expressed in the form of equation (5) as follows:

$$Y = \sum_{i=1}^{n} a_i V_i. \qquad \text{(D)}$$

The sum of the squares of the ordinates SSy is next computed and found equal to 11,503.664. When the $(SSR)_j$ for V_j $(1 \leqq j \leqq 7)$ is computed (Table I-B), it is seen that $(SSR)_2 = 11{,}500.848$ is the maximum. The desired form of equation (B), therefore, is as follows:

$$Y = a_0 + a_2 X^2. \qquad \text{(E)}$$

It is now necessary to compute the numerical values of a_2 and a_0. These values are computed by using equation (8) as follows:

$$a_2 = \frac{\Sigma\, y v_2}{\Sigma\, v_2^2}$$

$$a_2 = 170.86072$$

$$a_0 = \overline{Y} - a_2 \overline{V}_2$$

$$a_0 = 140.31875\text{-}(170.86072)\ (0.731)$$

$$a_0 = 15.41957.$$

The computed values of the a_i terms gives the following analytical function for the curve (Fig. 1) plotted from the original data:

$$Y = 15.41957 + 170.86072 X^2.$$

The variance $\sigma^2{}_{Y:X^2}$ must now be determined and may be computed by the following formula:

$$N\,\sigma^2{}_{Y:X^2} = S_y^2 = SSy - SSR$$

$$N\,\sigma^2{}_{Y:X^2} = 11{,}503.664 - 11{,}500.848$$

$$N\,\sigma^2{}_{Y:X^2} = 2.816.$$

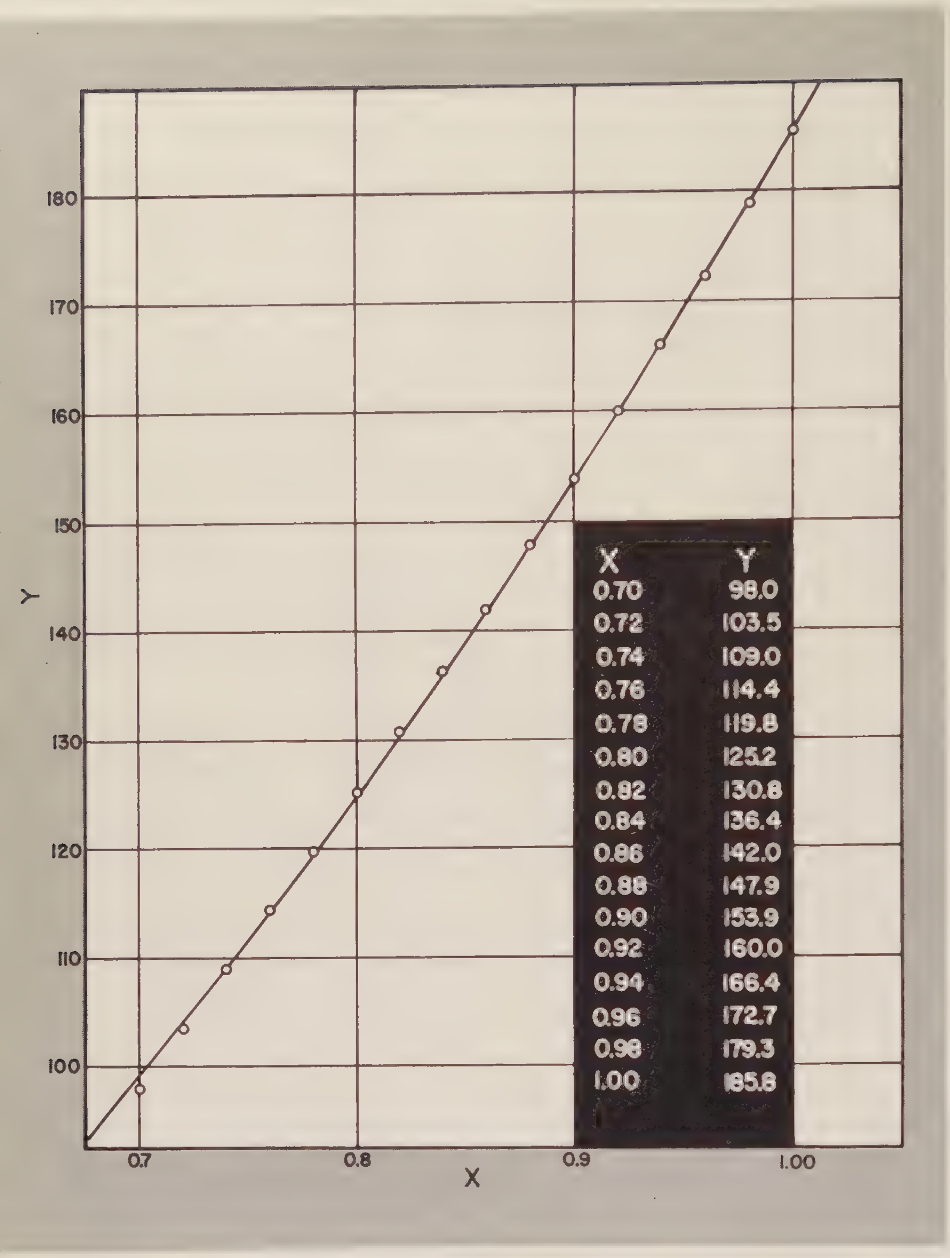


Fig. 1—The X and Y data points, when plotted, approximate a straight line. The method of least squares makes possible the determination of an equation for Y in terms of X which fits the plotted data.

A		B	
Parameter	Mean	Parameter	$(SSR)_j$
Y	140.31875	V_1	11,489.274
V_1	0.85000	V_2	11,500.848
V_2	0.73100	V_3	[illegible]
V_3	0.63580	V_4	11,372.531
V_4	[illegible]	V_5	11,241.556
V_5	[illegible]	V_6	11,073.895
V_6	[illegible]	V_7	[illegible]
V_7	[illegible]		

Table I—This table summarizes the results of calculations needed in the solution to example I.

If the variance is computed by the equivalent formula, however, the following result is obtained:

$$\mathcal{N}\sigma^2_{Y:X^2} = \sum^{n} (Y_{\text{observed}} - Y_{\text{calculated}})^2$$

$$\mathcal{N}\sigma^2_{Y:X^2} = 2.812.$$

The variance, therefore, equals

$$\sigma^2_{Y:X^2} = 0.176.$$

The standard deviation equals

$$\sigma_{Y:X^2} = 0.419.$$

The coefficient of correlation equals

$$r = \sqrt{(SSR)/(SSy)}$$

$$r = \sqrt{11{,}500.848/11{,}503.664}$$

$$r = 0.9999.$$

The difference between the variance values of 2.816 and 2.812 results from the numerical method of computation employed.

Example II

The second numerical example concerns determination of an analytical function to fit a specific set of data (Fig. 2) such that the maximum error in Y for any of the data points shall be less than 0.010.

The assumption is made that the following equation will satisfy the above criterion:

$$Y = \sum_{i=0}^{7} a_i X^i. \qquad \text{(F)}$$

First, the effect of the a_0 term is eliminated, and the term with the maximum $(SSR)_i$ determined. This determination shows that the X^4 term in equation (F) has the maximum $(SSR)_i$. The effect of this term is then removed and the absolute error at each of the 18 data points calculated (Table II-A). The calculations indicate that the term with the maximum $(SSR)_i$ to fit the error function is X^1. The effect of this term is removed and the absolute errors at the 18 data points again calculated (Table II-B). This calculation shows that the term whose $(SSR)_i$ is a maximum in fitting the error curve is X^7. The effect of this term is removed and the absolute errors at the data points again calculated (Table II-C). The term whose $(SSR)_i$ is a maximum in fitting the error curve of Table II-C is found to be X^6. Once again, the effect of this maximum $(SSR)_i$ term is removed and the absolute errors at the 18 data points

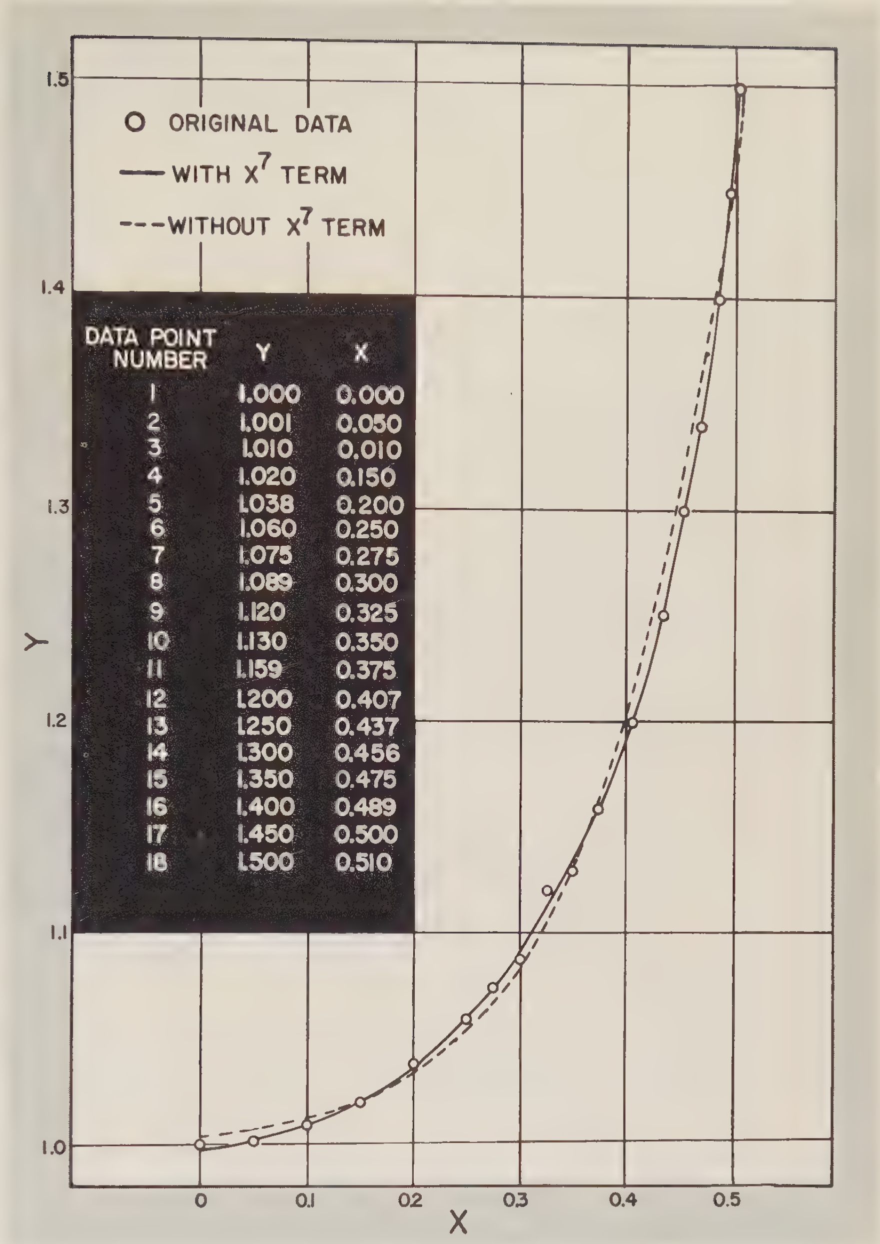


Fig. 2—The curves shown here were plotted from 18 data points. By using the method of least squares it is possible to determine an analytical function which will fit the data in such a way that the maximum error in Y for any of the data points will be less than 0.010. The significance of the 2 curves is as follows: (a) the more terms in the equation, the better the fit and (b) the number of inflection points on the curve may be increased as the number of terms in the equations is increased.

calculated (Table II-D). The last set of calculations show that the maximum error in Y for any of the data points is less than 0.010. This satisfies, therefore, the "goodness of fit" criteria and makes possible the determination of the numerical value of a_i by the use of equations (8), (10), and (6) as follows:

$$a_6 = \frac{\Sigma y v_6}{\Sigma v_6^2}$$

$$a_7 = \frac{\Sigma y v_7}{\Sigma v_7^2} - a_6\left[\frac{\Sigma v_6 v_7}{\Sigma v_7^2}\right]$$

$$a_1 = \frac{\Sigma\, y v_1}{\Sigma\, v_1^2} - a_7\left[\frac{\Sigma\, v_1 v_7}{\Sigma\, v_1^2}\right] - a_6\left[\frac{\Sigma\, v_1 v_6}{\Sigma\, v_1^2}\right]$$

$$a_4 = \frac{\Sigma\, y v_4}{\Sigma\, v_4^2} - a_1\left[\frac{\Sigma\, v_4 v_1}{\Sigma\, v_4^2}\right] - a_7\left[\frac{\Sigma\, v_4 v_7}{\Sigma\, v_4^2}\right] - a_6\left[\frac{\Sigma\, v_4 v_6}{\Sigma\, v_4^2}\right]$$

Table II—To obtain "goodness of fit" when determining an analytical function to fit a specific set of data, it is necessary to calculate the absolute errors at the various data points until the least possible error exists. This table summarizes the absolute error calculations for the 18 data points shown in Fig. 2 and indicates (D) that the maximum error in the dependent variable of the required equation for any of the data points is less than 0.010.

A		B		C		D	
Data Point Number	Absolute Error	Data Point Number	Absolute Error	Data Point Number	Absolute Error	Data Point Number	Absolute Error
1	0.020	1	0.003	1	0.010	1	0.002
2	0.019	2	0.007	2	0.001	2	0.002
3	0.011	3	0.003	3	0.005	3	0.000
4	0.004	4	0.001	4	0.009	4	0.000
5	0.007	5	0.006	5	0.006	5	0.002
6	0.013	6	0.008	6	0.003	6	0.000
7	0.016	7	0.010	7	0.000	7	0.000
8	0.014	8	0.007	8	0.000	8	0.004
9	0.024	9	0.016	9	0.014	9	0.008
10	0.008	10	0.000	10	0.003	10	0.004
11	0.005	11	0.003	11	0.005	11	0.001
12	0.006	12	0.013	12	0.001	12	0.001
13	0.015	13	0.020	13	0.004	13	0.000
14	0.013	14	0.016	14	0.002	14	0.004
15	0.015	15	0.016	15	0.008	15	0.001
16	0.008	16	0.005	16	0.006	16	0.002
17	0.006	17	0.011	17	0.001	17	0.000
18	0.021	18	0.028	18	0.009	18	0.001

Example III

The third numerical example concerns determination of an analytical function $Y = f(X,X_2)$ for a specific set of data (Table III).

In compliance with basic assumptions, the function $Y = f(X_1,X_2)$ is first expressed as follows:

$$Y = G_1(X_1)\,G_2(X_2).$$

A polynomial fit of nothing higher than the seventh degree for each of the functions $Y = G_i(X_i)$ is assumed, and preliminary runs are made to determine the predominate terms in each of the $G_i(X_i)$. The first 4 predominate terms in $G_1(X_1)$ for $X_2 = 1.03$ are X_1, X_1^2, X_1^3, and X^7, respectively. The first 4 predominate terms in $G_1(X_1)$ for $X_2 = 1.11$ are X_1, X_1^2, X_1^3, and X_1^4 respectively. The first 4 predominate terms in $G_1(X_1)$ for $X_2 = 1.28$ are X_1, X_1^2, X_1^3, and X_1^4. With this information the following equation for $G_1(X_1)$ can be written:

$$G_1(X_1) = b_0 + b_1 X_1 + b_2 X_1^2 + b_3 X_1^3 + b_4 X_1^4.$$

Since the data for $G_2(X_2)$ for a fixed X_1 are available at 3 points, only the first most predominate term for 3 different values of X_1 is found. The most predominate term in $G_2(X_2)$ for 3 different values of X_1 is as follows:

X_2 for $X_1 = 1.30$
X_2 for $X_1 = 1.40$
X_2^2 for $X_1 = 1.50$.

Y	X	X_2
29.1	1.150	1.03
32.1	1.200	1.03
34.3	1.250	1.03
35.8	1.300	1.03
36.8	1.350	1.03
37.6	1.400	1.03
38.1	1.450	1.03
38.5	1.500	1.03
38.8	1.500	1.03
24.3	1.150	1.11
28.5	1.200	1.11
31.4	1.250	1.11
33.4	1.300	1.11
34.8	1.350	1.11
35.8	1.400	1.11
36.5	1.450	1.11
36.8	1.475	1.11
37.1	1.500	1.11
37.4	1.550	1.11
37.6	1.575	1.11
37.7	1.600	1.11
24.5	1.250	1.28
28.2	1.300	1.28
30.6	1.350	1.28
32.0	1.400	1.28
33.0	1.450	1.28
33.8	1.500	1.28
34.4	1.550	1.28
35.0	1.600	1.28

Table III—The method of least squares can be applied to determine an analytical function for Y in terms of X and X_2 which will fit the specific set of data summarized in this table.

From the above information, though one must realize its incompleteness, the following equation for $G_2(X_2)$ can be written:

$$G_2(X_2) = C_0 + C_1 X_2 + C_2 X_2^2 + C_3 X_2^2 + C_4 X_2^4.$$

The relationship $f(X_1, X_2) = G_1(X_1)\,G_2(X_2)$ can now be determined as follows:

$$\begin{aligned} f(X_1, X_2) = {} & a_0 + a_1 X_1 + a_2 X_1^2 + a_3 X_1^3 \\ & + a_4 X_1^4 + a_5 X_2 + a_6 X_1 X_2 \\ & + a_7 X_1^2 X_2 + a_8 X_1^3 X_2 + a_9 X_1^4 X_2 \\ & + a_{10} X_2^2 + a_{11} X_1 X_2^2 + a_{12} X_1^2 X_2^2 \\ & + a_{13} X_1^3 X_2^2 + a_{14} X_1^4 X_2^2 + a_{15} X_2^3 \\ & + a_{16} X_1 X_2^3 + a_{17} X_1^2 X_2^3 + a_{18} X_1^3 X_2^3 \\ & + a_{19} X_1^4 X_2^3 + a_{20} X_2^4 + a_{21} X_1 X_2^4 \\ & + a_{22} X_1^2 X_2^4 + a_{23} X_1^3 X_2^4. \end{aligned}$$

After the effect of $\overline{Y}$ is removed, the sum of the squares of the errors (*SSE*) is found to equal 450.60, and the maximum error at any of the original data points is 9.6. The most predominate term then is determined to be X_1. After the effect of this term is removed, the *SSE* is found to equal 204.03 with a maximum error of 6.4. The most predominate term is then determined to be $X_1^3 X_2$. After removing the effect of this term, the *SSE* is found to equal 53.37 with a maximum error equal to 3.8. The most predominate term then is found to be $X_1^4 X_2^2$. The effect of this term is removed, and the *SSE* is found to equal 16.78 with a max-

imum error of 1.7. The most predominate term then is determined to be X_1^2. After the effect of this term is removed, *SSE* is equal to 15.0 with a maximum error of 1.8. (It is to be noted that there has been a very slight reduction in the *SSE* while the maximum error at 1 of the data points has increased. This situation can be explained by considering the geometric orientation of the axes.) Further analysis shows that the most predominate term is X_2. The effect of this term is removed and the *SSE* found equal to 5.68 with a maximum error equal to 1.3. The most predominate term next is determined to be X_1^3. After removing the effect of this term, the *SSE* equals 3.77 with a maximum error of 1.0. The most predominate term then is determined to be X_1X_2. The effect of this term is then removed, and the *SSE* found to equal 0.55 with a maximum error of 0.2.

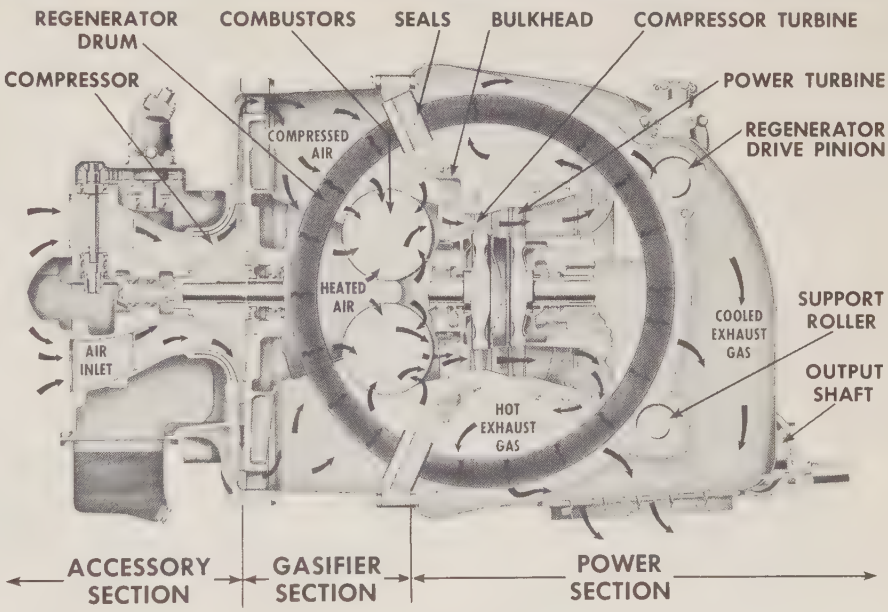
The use of a subsequent term will not reduce the maximum error in the ordinates. All additional coefficients, therefore, can be set to equal zero. After the numerical value of the coefficients is determined, the final equation of the required analytical function can be expressed as follows:

$$Y = -159.46481 + 907.58692X_1 - 915.5674X_1^2 + 272.06073X_1^3 - 463.98795X_2 + 439.07302X_1X_2 - 64.722750X_1^3X_2 + 0.52484X_1^4X_2^2.$$

Summary

The development of this new method of curve fitting has made it possible to fit an analytical equation using more than 1 independent variable. The method has proven very useful in extrapolating the operation characteristic data of a turbo-prop control system. The advantage of the new method (reducing the number of terms in an equation) has allowed replacing large tables of data by a simple equation to study the characteristics of a new engine by using high-speed electronic computing equipment. It also will be possible to use this new method in extending business cycle data to determine the significance of a given parameter.

GM Engineers Develop Experimental Gas Turbine Powered Truck



Through the cooperative efforts of Chevrolet Motor Division and GM Research Staff engineers has come the development of an experimental gas turbine powered truck, called the Turbo-Titan. The Turbo-Titan combines a stock Chevrolet heavy-duty, tandem-axle tractor chassis (above) with a modified version of the GM Research Staff GT-304 Whirlfire gas turbine engine (below). The GT-304, originally developed for the experimental Firebird II passenger car, is mounted in place of the usual piston-type engine and is coupled to a conventional power train having a slightly modified automatic transmission.

The 200 hp, air-cooled, Whirlfire gas turbine engine weighs approximately 850 lb and has a fuel-saving regenerator which recovers 80 per cent of the heat in the exhaust gas. The engine has 3 primary sections: (a) the accessory section, (b) the gasifier section, and (c) the power section (right).

The gasifier section of the GT-304 is simply a shaft with a single-stage, centrifugal-type air compressor on one end and a single-stage, axial flow-type turbine wheel on the other. At rated full power, the air compressor and the compressor turbine wheel rotate at 35,000 rpm. Idle speed is 15,000 rpm. Compressor ratio is 3.5 to 1. The stream of hot exhaust gas from the gasifier section is used to turn the turbine wheel in the power section. The power turbine, with a rated turbine speed of 28,000 rpm, is connected through reduction gears to the truck transmission.

The regenerator drum is of metal mesh construction and rotates at approximately one-thousandth the speed of the gasifier shaft. The drum rotates first through the hot exhaust gas and then through the relatively cool compressor discharge air, transferring heat from the exhaust gas to the incoming air. Since the temperature of the incoming air is raised mostly by heat supplied by the regenerator, only enough fuel need be burned to raise the pre-heated inlet air to the normal operating temperature of 1,650° F at the turbine inlet.

Extensive highway and proving ground tests have shown the Turbo-Titan to have exceptional hill climbing ability as compared to a similar vehicle powered by a piston-type engine of the same horsepower rating. This reflects the superior torque characteristics of the gas turbine engine. Other advantages cited for the gas turbine engine include low maintenance requirements, excellent cold weather starting characteristics, no need for warm up, wide choice of fuels, and the elimination of oil changes since the oil is not contaminated by combustion products.

The Turbo-Titan gas turbine truck is not planned for production at the present time. Developmental work is of an experimental nature for the purpose of investigating the potential of the gas turbine truck in the heavy-duty commercial field.

Industrial Engineers at Work:

Some Typical Processing Problems Resulting from an Annual Model Change

BUICK
OLDS
PONTIAC

To many people the glamour phase of the automobile business is the final assembly operation which, to them, represents the entire process involved in manufacturing a car. What is not known or thought about is the behind-the-scenes activity started many months before a new car model is ready for assembly. This activity is devoted to the solution of many problems resulting from a new model change. Methods, process, and plant engineers must thoroughly analyze what effect such items as new methods of assembly, new tooling required, and new parts will have on plant layout, material handling and stocking, and scheduling. The solution to the many problems involved does not stem from one engineer or one group of engineers. They are solved as the result of close teamwork between all engineers concerned, each of whom contributes his share of knowledge and experience to the ultimate goal—the final assembly of an automobile.

THE annual model change program carried out by the American automotive industry introduces new ideas in engineering design and styling. At the same time, the annual model change introduces new processing problems to engineers engaged in various phases of assembling the new model. The development of new methods of assembly and

Fig. 1—On the final assembly line at the Kansas City, Kansas, plant of Buick-Oldsmobile-Pontiac Assembly Division, 3 different makes of passenger cars are assembled on a common assembly line. Other B.O.P. assembly plants are located at Arlington, Texas; Atlanta, Georgia; Linden, New Jersey; South Gate, California; Framingham, Massachusetts; and Wilmington, Delaware.

By JOSEPH M. SHERWOOD
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Buick-Oldsmobile-Pontiac
Assembly Division

The engineering required after a car is designed and before it is assembled

tooling and changes in established procedure make it necessary for methods, process, and plant engineers to analyze carefully how the model change will affect their particular area of activity. What new facilities are needed? What facilities must be altered? Where will additional assembly line length be needed? How much additional storage area is needed for new parts? What new methods of assembly are required? These are a few of the typical questions which must be answered prior to finalizing a model change program.

Processing problems relating to the assembly of a single make of car are many. The assembly of 3 different makes of passenger cars on a common assembly line (Fig. 1), as carried out by the Buick-Oldsmobile-Pontiac Assembly Division, causes the problem to be expanded. At each of the 7 B.O.P. plants, the 3 makes of cars are assembled from parts designed and fabricated by other GM Divisions and outside suppliers. The cars are assembled according to strict specifications of each car Division and Fisher Body Division. These Divisions also furnish all tooling required for building of the cars.

The annual model change in a B.O.P. assembly plant starts with obtaining information regarding (a) the separate items of major tooling to be used and the number of different car models to be built from this tooling, (b) the difference between the new method of assembly in relation to the present method, (c) the number of material items involved in the new method of assembly, (d) the method for delivering sub-assemblies built by the new tooling to the point of use, (e) size of the new tooling, material, and sub-assemblies, (f) additional and new operations required to build the new body or bodies, and (g) new equipment to be purchased for the new operations. Some of the required information is obtained from the B.O.P. Central Office, located in Detroit. The remaining information can be obtained only after a study of a particular B.O.P. assembly plant's facilities as they relate to the changes. In some cases the information obtained is such as to cause a major rearrangement of an entire assembly department.

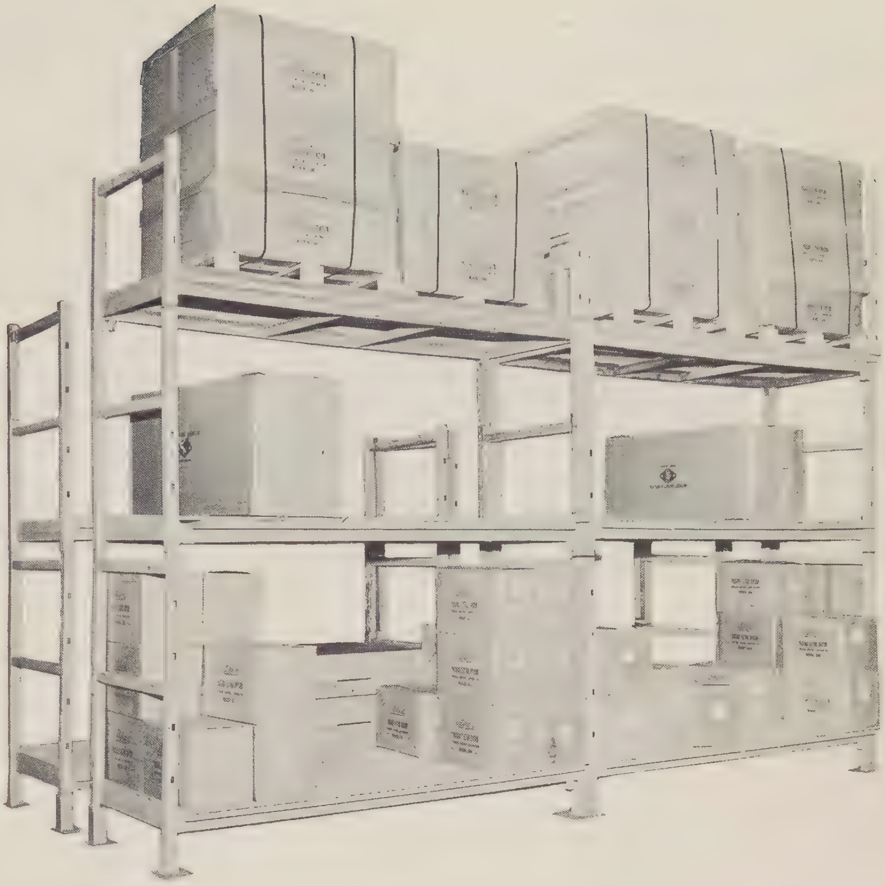


Fig. 2—Universal, tiering storage racks, such as those shown here, are used in a B.O.P. plant to store the more than 15,000 separate parts used in the assembly of Buick, Oldsmobile, and Pontiac passenger cars. The storage racks must be versatile to allow for parts stocking near the assembly lines.

New Assembly Methods Can Create Material Stocking and Delivery Problems

With each model change, new methods of assembly are introduced as a result of new tooling used to simplify the construction of a body and provide a more efficient assembly operation. New methods of assembly, however, often cause rearrangement of a major sub-assembly area which, in turn, creates problems of material stocking, delivery to the assembly line, and area requirements for the placement of stock.

To obtain the required stock area, 4 factors must be considered: (a) area required, (b) area layout best suited for the most efficient stocking operation, (c) accessibility of the area for stocking by mechanical handling equipment, and (d) ability of the area to handle all stock, including back-up stock. If back-up stock is to be located elsewhere, the same factors again must be considered.

To provide the necessary stock area, one or more assembly conveyor systems often must be moved and positioned either overhead or underground to allow a crossover for mobile equipment. In some cases short, quick-transfer conveyors are used. These conveyors do not cause as much usable line length loss as do the inclines and declines of the overhead and underground conveyors.

Movement of assembly lines into an area previously used for back-up stock can create another problem—where will

Fig. 3—Three body shells, as used in certain models of 1957 Buick (A), Oldsmobile (B), and Pontiac (C) passenger cars, are shown in a body checking area. The Pontiac body shell is shown positioned on a "dry dock" of a dimensional checking bridge. This equipment is used to make an accurate check of body shell dimensions.

the back-up stock be placed? To take care of the reduction in available floor area and provide for the assumption that an equal, if not larger, volume of material will have to be handled in the new model year, special tiering stock racks are used (Fig. 2). Since 3 different makes of cars are assembled in a B.O.P. plant, the tiering stock racks are universal in design to handle material for all 3 car makes.

Relocation of assembly lines presents a different problem from that of material stocking. The governing factor in relocation of assembly lines is that they must be set up to conform to the new method of assembly.

Welding Operations for the Different Types of Bodies Are Combined Where Possible

Welding constitutes the major operation performed in the assembly of various automobile body components. Of the 3 welding methods used at B.O.P.—resistance, gas, and arc—the resistance spot welding method is used approximately 90 per cent of the time because of its speed, ease, and versatility.

The bodies used for each of the 3 different makes of cars (Fig. 3) require different sub-assembly components and welding operations. The different components require different clearances, different tooling for holding and locating parts, and different spot-weld tools specified for welding the components and assemblies. Although components are different, the majority of welding operations for each body are similar in nature.

The different components used on each body cannot be changed to make universal parts. Also, the expense involved in making all locating and holding tools universal would be prohibitive. To limit the cost of assembly by effective utilization of manpower and assembly line length, therefore, similar welding operations are grouped on the different body types. This is done by combining the spot-weld tooling for the similar operations to arrive at universal spot-weld tools for use on all 3 makes of cars.

Combining spot-weld tooling for similar welding operations is a long and complicated process. Experience is the biggest asset to be had. The process is aided, however, by Fisher Body "pilot programs," which make it possible to witness the actual welding operations required to assemble the 3 makes of cars. Knowledge gained from the pilot program makes it possible to check out assumptions regarding what spot-weld tools are to be used in the new method of assembly and if the operational sequence and assembly line length are conducive to producing a quality product.

All welding sub-assembly areas are handled in the same manner, since each sub-assembly is considered as a miniature, independent assembly line. After the welding operations are organized, the next step is to set up the assembly line on which the operations are to be performed.

Welding operations pertaining to body assembly are specified on a series of blueprints, called *welding studies*, supplied by Fisher Body. These prints specify weld location, number of spots or length of bead, and the welding tool required (Fig. 4). The welding studies are different for all body styles although for a particular car make, they are quite similar.

The welding studies are useful in combining similar operations for all car makes and body styles in order to condense assembly line space, equipment, and tooling. To accomplish this, information on welding operations required for a particular body style is transposed from the welding study to tracing paper —that is, individual spot welds for a given operation, represented by an *X* on the welding study, are marked on a large sheet of tracing paper. This same sheet of tracing paper is used for each different welding study. After all spots are marked on the tracing paper, a picture of the welding problem is formed. Each operation to be made by a particular spot weld tool is designated by a specific color. This makes it possible to determine the number of separate opera-

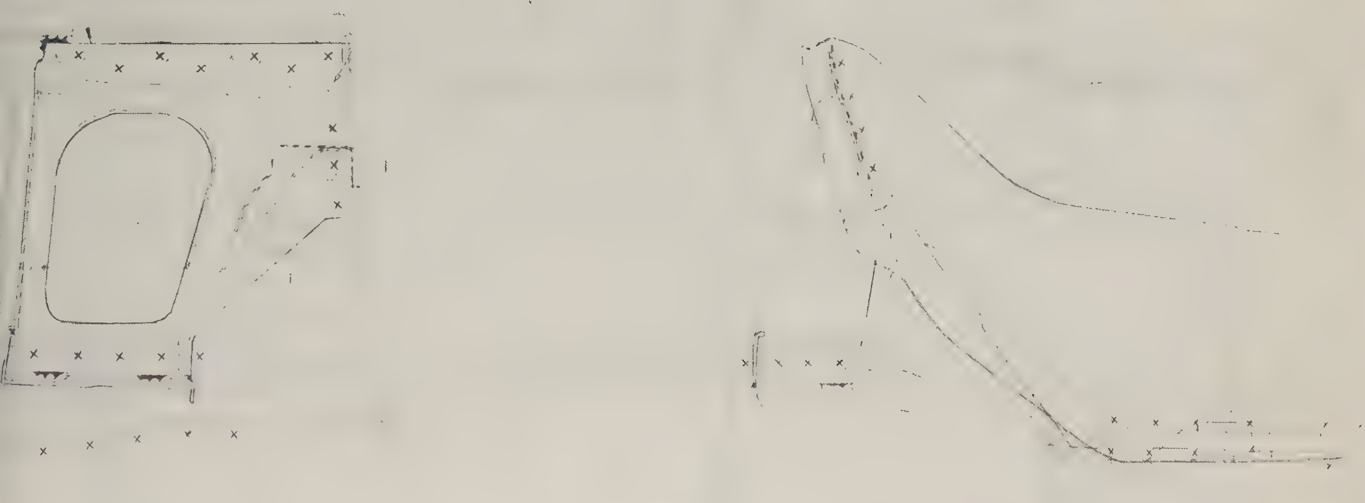


Fig. 4—Shown here is a photograph of a typical *welding study* supplied to B.O.P. by the Fisher Body Division. This particular welding study is for a dash-to-chassis frame brace spot-welded to a toe brace sub-assembly. The spot-welding gun is shown in welding position. The X's indicate the location and number of spot welds desired. The welding studies are used to determine the number of separate spot-welding operations required by a single tool for all bodies assembled at a B.O.P. assembly plant.

tions required by this tool for all bodies. This information is then used to specify the complete operational sequence and assembly line length, the number of spot-weld tools required, and the number of combination or special spot-weld tools which must be made (Fig. 5).

Model Change Also Causes Processing Problems in Paint and Trim Shops

Processing problems usually do not occur in the Paint Shop of a B.O.P. plant except in the case of a change in painting procedure. For example, one component of the 1957 models, which in 1956 was installed on a finished, painted body, was specified to be received in the Paint Shop in a primed condition, installed on a primed body, and then painted body color. This required the provision of stock area and assembly line length necessary to install the part on the body. However, the required area and line length did not exist. It became necessary, therefore, to obtain the required area from material storage and to rearrange conveyors to provide the assembly line length needed. The common assembly line for the 3 car makes caused this problem to be expanded, since the change affected all models. Also, because the component had a different design for each body style, more area and line length were required than would have been the case for a single car-make assembly line.

The majority of processing problems in the Trim Shop (to which the body is conveyed after leaving the Paint Shop to receive items to trim the interior, openings, and lines) stem from rearrangement

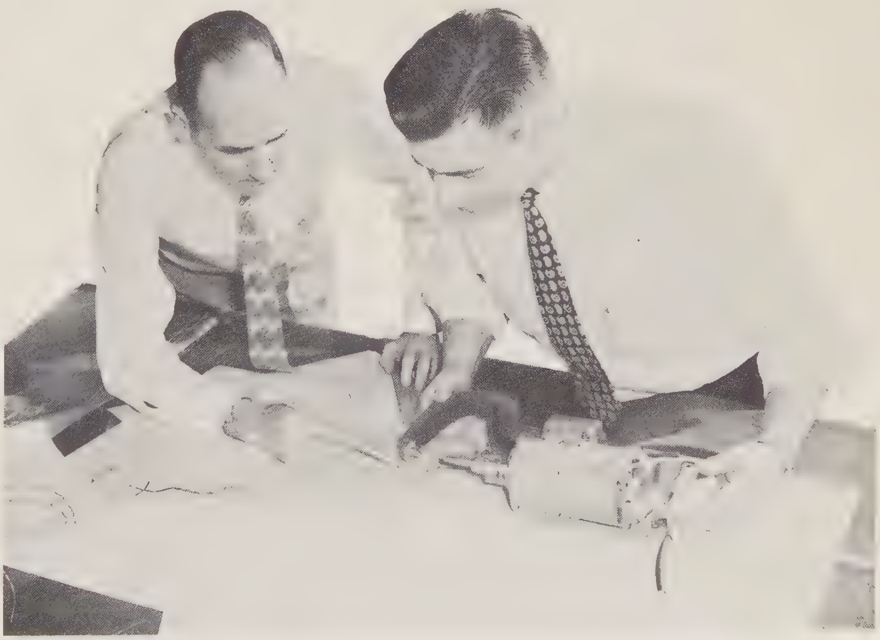


Fig. 5—Shown here are B.O.P. process engineers as they work from 3 different views of 3 separate welding guns to design a single welding gun which will perform the welding operations required by the 3 separate guns.

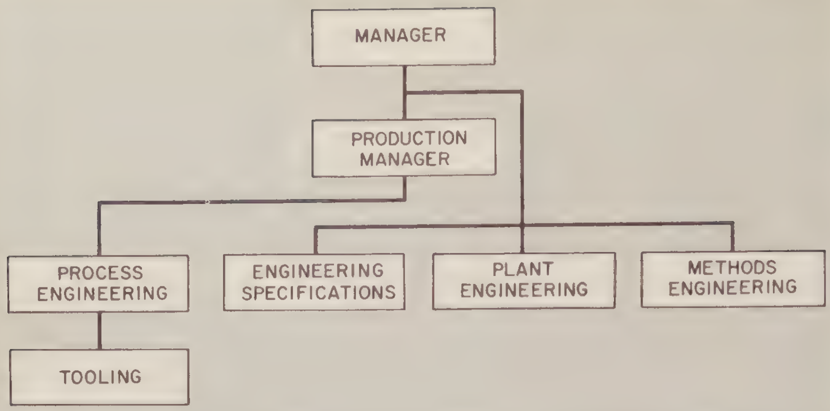


Fig. 6—This chart shows the organizational arrangement of the engineering Departments at the B.O.P. Kansas City plant and the manner in which each Department reports to the plant manager.

of assembly operations and stock areas. For example, the 1957 Buick and Oldsmobile bodies required a new method of assembly due to the use of a 3-piece, rear window glass in place of the previously used 1-piece glass. The new method meant tripling the amount of on-the-line stock, sub-assembly area, and sub-assembly bank area. Because the rear window glass assembly area is one of the most important and difficult areas to move in the Trim Shop, it was necessary to rearrange assembly activities around this area by relocating conveyors, changing assembly sequences, and transferring operations.

Occasionally, a Trim Shop processing problem results from the installation of a new assembly facility. For example, on the 1957 Buick and Oldsmobile bodies, the dash panel was redesigned in such a way that it prevented the usual practice of placing the body on a chassis with the steering column already installed. This change required the steering column to be installed on the body before the body was placed on the chassis. Because the body is placed on the chassis after it leaves the Trim Shop, the steering column installation facility had to be located in the Trim Shop. This required considerable stock working area and bank. The rearrangement resulting from the addition of the steering column installation facility required obtaining a length of assembly line to provide operational room.

Engineers Work as Co-Ordinated Team When Planning Model Change Program

The discussion so far has dealt with some of the processing problems encountered by B.O.P. engineers resulting from a model changeover. But what of the overall plant organization behind the planning necessary to affect the changeover? How does it function? What are the various departments, both engineering and non-engineering, which combine their talents to achieve a smooth changeover relatively free from major problems? The Kansas City, Kansas, plant of B.O.P. will serve to illustrate the function and responsibilities of various engineering departments (Fig. 6).

It has been the experience of B.O.P. to anticipate processing problems before they develop. For this reason, a permanent Model-Change Committee, which reports directly to the plant manager, coordinates all planning. The Committee, composed of representatives from the Specifications, Plant Engineering, Material, Processing and Tool Design, Work Standards, and Methods-Layout Departments, normally begins operation shortly after the previous model change is completed. Other members of the Committee, who advise on specific problems, are personnel of the Accounting, Scheduling, Traffic, and Purchasing Departments and Production Department supervisors.

All Departments comprising the Model Change Committee do not work as a group on all problems. For example, a specific problem relating to a new assembly method would be worked on only by the Processing and Tool Design and Methods-Layout Departments.

Engineering Specifications Department

The Engineering Specifications Department obtains all new-model product engineering and specifications data from the 3 car Divisions and Fisher Body. This information is then routed to all Departments concerned and also to the Model-Change Committee.

The Engineering Specifications Department also has responsibility for follow-up to make sure that separate assemblies and the total product are built according to the car Divisions' specifications. Also, last minute changes, necessitating some rework of material, require the Department to provide the necessary information and a follow-up to see that the rework is done according to specifications. This Department, due to its function, works closely with the Detroit B.O.P. Central Office.

Material Department

The Material Department is responsible for receiving, storing, and final delivery of material to assembly line stations. There are about 15,000 different pieces of material for a car, each requiring some type of special handling method. These methods must be checked because of changes in parts dimensions for the new model year.

Even though standard parts are used whenever possible in assembling each of the 3 makes of cars, new parts are numerous and must be checked for shipping method, packaging, handling facilities required, how and where to store, and how and where the parts are to be placed for the assembly line operator.

For example, the engine, which represents a new part in a model change, is shipped to a B.O.P. plant in railroad boxcars as a completely assembled unit, except for minor attachments. The Material Department must first analyze the size, weight, and contour of all engines to be received so that the number of unloading hooks required can be decided upon. The Department is then concerned with the different types of engines to be received per load so that the necessary unloading dock facilities will be available. The Department also must consider the storage aspect of the engines. In this regard, existing storage racks used to store the previously used engines must be revised to handle the new engines (Fig. 7). Detailed planning of this nature is carried out for each of the parts to be handled by the Material Department.

Work Standards and Methods-Layout Departments

The Work Standards Department plays an integral part in the operational planning for model change by making a thorough analysis of each new operation to determine how it can be added to a

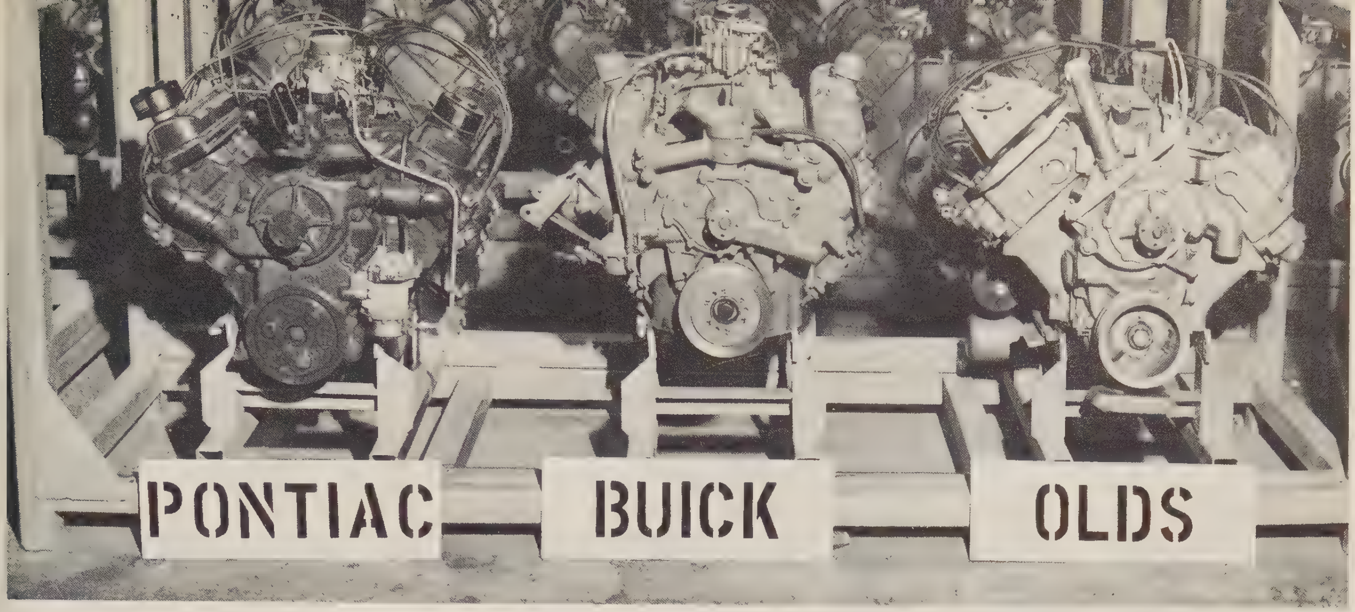


Fig. 7—Each model change requires the revision of existing engine storage racks to handle new engine designs. The storage racks shown here are designed to handle engines for each make of car assembled at a B.O.P. plant. Tiering racks are used to reduce the overall number of racks required.

carry-over operation to form balanced operations. The new operations are planned in cooperation with engineers from the Production Control and Specifications Departments. After the operations are finalized, operator work assignments are established for assembly line operations.

The Kansas City B.O.P. plant has the Methods and Layout Departments, usually separated, combined into one Department which has responsibility for providing effective use of manpower and facilities. After the pre-model change plan has been established, the Department's engineers make a thorough analysis and detail layout of all assembly operations. Requirements set up by the Specifications, Processing, and Work Standards Departments are first studied. Plans then are finalized after meetings with engineers from the Production and Material Departments. After all detail planning is complete, the Plant Engineering Department is given layouts for detailing the necessary assembly facilities.

Engineers of the Methods-Layout Department also must consider trends of assembly from year to year. The length of an assembly line conveyor formerly was established by the number of assembly operations. Now, the amount of material required in a specific assembly operation is the dominant factor. For example, trim combinations have increased from 37 in 1950 to 225 in 1957, Basic body colors have increased from 39 to 54 for the same period. The amount of material required to supply this increase has not only lengthened the assembly line but also has made the provision of storage area a difficult problem. The area must be large enough to store the different trim items and also be accessible so that each trim and color combination can be reached easily by material handling equipment.

Methods-Layout engineers also must make sure that material is placed within a reasonable distance of the assembly line operator. The stock area required for one assembly line operator has increased in the last 5 years. To reduce facilities expansion, a method has been developed whereby material, required for a day's run, is moved by mobile equipment from a bulk storage back-up area to the assembly line on a pre-determined, approximate schedule. The schedule is the result of experience and experimentation. When it is economical, conveyors are used to carry material from bulk storage into a crowded assembly area.

Plant Engineering Department

During the planning stage of the model changeover, the Plant Engineering Department determines what new assembly facilities are required or what existing facilities must be rearranged for the overall model change program. This Department also does the estimating and preparation of an appropriation request for approval by the B.O.P. Central Office.

The layouts submitted to the Plant Engineering Department, after all detailed planning is complete, are the basis for detailed structural, electrical, and ventilation design of assembly facilities. The completed designs are then sent to the Maintenance Department for construction. All construction of facilities is planned for completion on a definite schedule. This schedule is based on the day that the new models are to be displayed in dealer showrooms.

Conclusion

The American automotive industry makes a practice of investing regularly in planned change. The annual model change is the industry's pattern of replacing the old with the new and allows for new ideas both in styling and engineering design.

The annual model change, however, presents many processing problems in the assembly of an automobile. At B.O.P. Assembly Division, where the problems encountered serve to stimulate the progressive thinking of the entire organization, efficient assembly depends not only upon the process or tooling used but also upon the harmonious cooperation of all engineers engaged in planning for the model change. Lengthy periods of down time no longer occur even though changes from one model year to the next are far greater now than they were in the past.

The annual model change means problems—but also progress. B.O.P. engineers meet the change with optimistic expectancy. Today, while the assembly lines at each of the 7 B.O.P. plants are in full production on 1957 models, plans are well underway for solving processing problems dealing with assembly of 1958 models.

The Cadillac Tubular Center X-Frame: A New Concept in Automotive Design

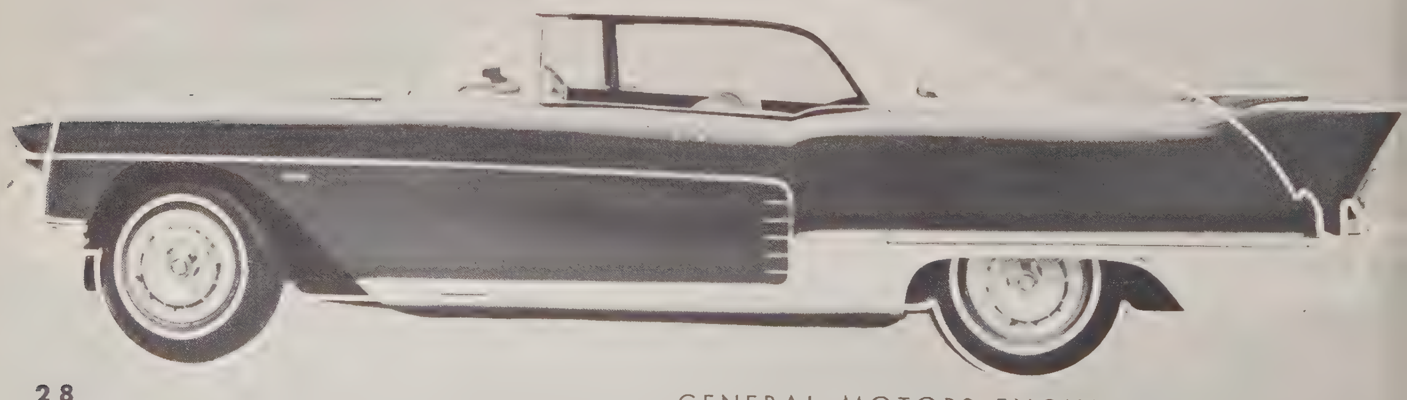
The 1957 Cadillac tubular center X-frame represents a new concept in automotive design for several important reasons. For years, the trend in automotive design has been toward lower cars for improved appearance and customer appeal. These designs, however, have often resulted in an accompanying trend toward slightly reduced structural efficiency, reduced passenger space, or both. The 1957 Cadillac frame design is distinctive in that it continues the trend toward a still lower car silhouette but reverses the trend toward reduced passenger space and reduced structural efficiency. This is accomplished by a frame design which essentially is a true "X," eliminating the conventional side rails of previous designs and utilizing a properly designed center section. The design was an evolutionary one, resulting from studies and development both by Cadillac Motor Car Division and by the frame manufacturer. It has the additional advantages of passenger safety because of lower center of gravity, adaptability to several wheelbase models, and acceptable fabrication process.

Fig. 1—The trend toward lower cars is emphasized by a comparison of Cadillac cars of several model years. The car outlines (right) represent some of the changes in styling that have occurred over a 19-year period. As each change was made to progressively lower cars, the requirements of the frame and body became more demanding. Roof rails and heavy pillars have been gradually reduced or eliminated. More glass areas have been used. Additional weight is imposed on the structure as a result of more glass, more conveniences, and more accessories on today's car. The 1938 model shown (above) was a Series 60 Special which was 2½ in. lower (at 65 in.) than the other Cadillac models of that year. Its outline is compared to the new Cadillac Eldorado Brougham (below) which has an overall height of 55½ in. All 1957 model Cadillac cars are lowered still further from 1956 cars, and the structural and space problems were solved successfully by the use of the new tubular center X-frame.

In automotive engineering, a new design to satisfy customer demand very often necessitates a compromise in some of the other features of the car. Lowering the overall height of a car usually brought about such compromises when new models were introduced in the past. Sacrifices in the form of increased weight, less rigidity of the vehicle or reduced passenger space have occurred in some of the previous designs when a lower body style has been introduced.

The development of the Cadillac tubular center X-frame, however, permitted a still lower car silhouette for 1957 models but allowed engineers to convert some of the expected compromises into improvements both in frame design and vehicle design.

The problems of how to lower car heights are not new for there has been a continuing trend in this direction for many years in the automotive industry. It is of interest to examine this trend and the developments which have occurred, for these had an important effect on establishing the requirements in the design of the 1957 Cadillac.



By LESTER MILLIKEN
Cadillac Motor Car
Division

In an American production car, a true X-frame for lower height, increased safety

Customer Preference Reflected in Trend to Lower Cars

The trend toward lower car styling has persisted consistently throughout the entire history of automotive progress as roads, materials, and automobile designs have improved. It has been said that car height reductions have averaged ½ in. per year since 1927.

Cadillac cars, of course, have been a part of this trend. In 1938, for example, the 60 Special was truly a "special" car for, at 65 in. high, it was 2½ in. lower than other Cadillac cars of that year. The new styling made possible by this marked reduction in height was well received and established the direction in car heights in future designs. With each new design following that year, a lower car was introduced, and the 1957 Cadillac Eldorado Brougham represents the lowest to date with a height of 55½ in. (Fig. 1).

The general trend has been continuous and definite. The few experiments in attempting to stabilize the height of cars have shown that the buying public does not feel that the added chair height and head room possible in a higher car is as desirable as the lower and longer silhouette. The constantly increasing millions of cars which have been built to the lower silhouettes by all manufacturers is certain proof that this is one phase of providing a more attractive product for the user. With the objective so definitely established, the engineer's continuing job has been to provide a frame and a chassis which would accommodate the lower bodies. Many different approaches have been made to the problem, and hundreds of experimental frames have been built and tested.

Cadillac designs for an integral frame

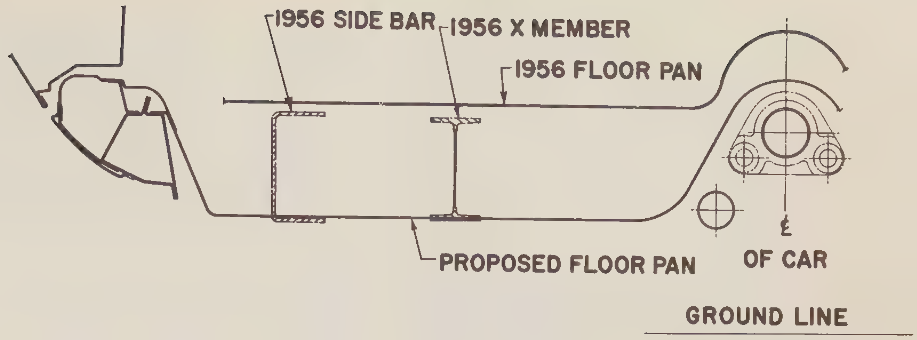
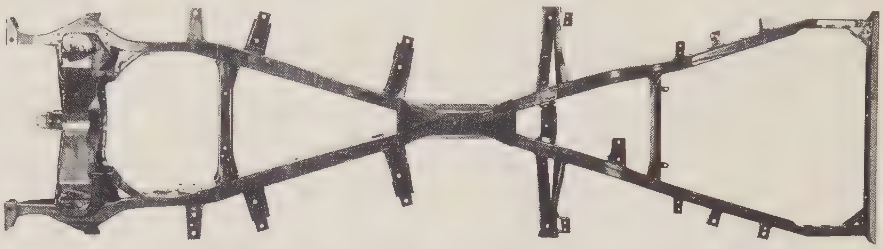


Fig. 2—The proposed floor-pan cross-section was studied in comparison to the 1956 frame design, and it showed that the 1956 frame would not be satisfactory for a lower body without sacrificing passenger space. The above diagram indicates the position of the 1956 frame side rail and X-brace in relation to the desired floor pan cross section for the Eldorado Brougham, which has lowest design condition of 1957 models.



STRUCTURAL PERFORMANCE DATA

FRAME AND COMPLETE CAR COMPARISON							
CAR NAME		COMPLETE CAR		FRAME			
YEAR AND BODY	TYPE	TORSIONAL RIGIDITY	MAX BEND DEFLECTION	WEIGHT	TORSIONAL RIGIDITY	MAX BEND DEFLECTION	
1957 CADILLAC	SEDAN DE VILLE	5560 LB-FT/DEG	0.047 IN.	410 LB	2900 LB-FT/DEG	0.102 IN.	
1956 CADILLAC	SEDAN DE VILLE	5200 LB-FT/DEG	0.084 IN.	393 LB	2700 LB-FT/DEG	0.144 IN.	
1956 CADILLAC	4 DOOR SEDAN	5500 LB-FT/DEG	0.062 IN.	369 LB	2590 LB-FT/DEG	0.160 IN.	
1956 CAR "A"	4 DOOR SEDAN	5300 LB-FT/DEG	0.050 IN.	330 LB	1860 LB-FT/DEG	0.171 IN.	
1956 CAR "B"	4 DOOR SEDAN	4160 LB-FT/DEG	0.047 IN.	325 LB	1450 LB-FT/DEG	0.152 IN.	
1956 CAR "C"	4 DOOR SEDAN	4750 LB-FT/DEG	0.031 IN.	377 LB	1880 LB-FT/DEG	0.121 IN.	

Fig. 3—This is the resulting basic design of the X-frame with the tubular center section. The extended brackets shown are the 4 major mounting locations for a separate body. The table summarizes the gains achieved in structural efficiency in comparison with 1956 Cadillac cars and a selection of other cars of about the same size.

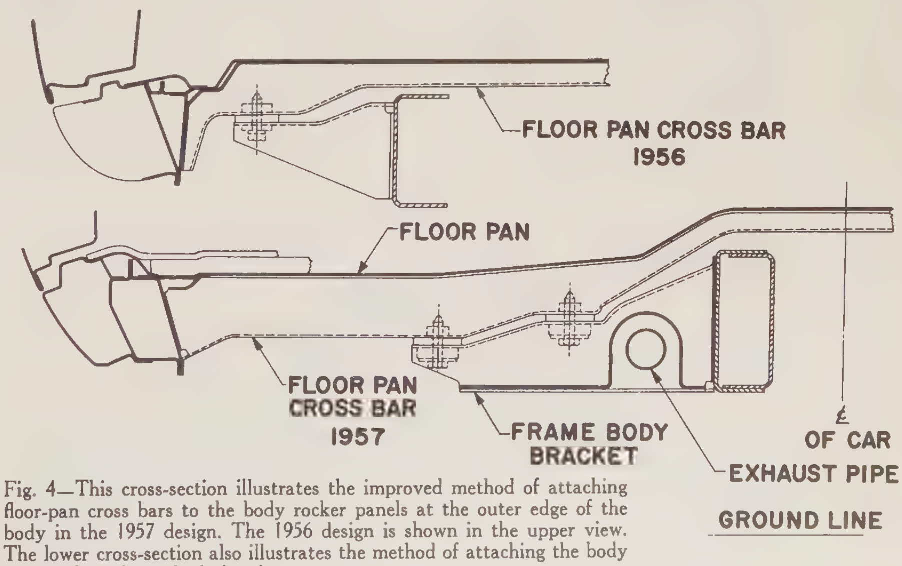


Fig. 4—This cross-section illustrates the improved method of attaching floor-pan cross bars to the body rocker panels at the outer edge of the body in the 1957 design. The 1956 design is shown in the upper view. The lower cross-section also illustrates the method of attaching the body to one of the frame body brackets.

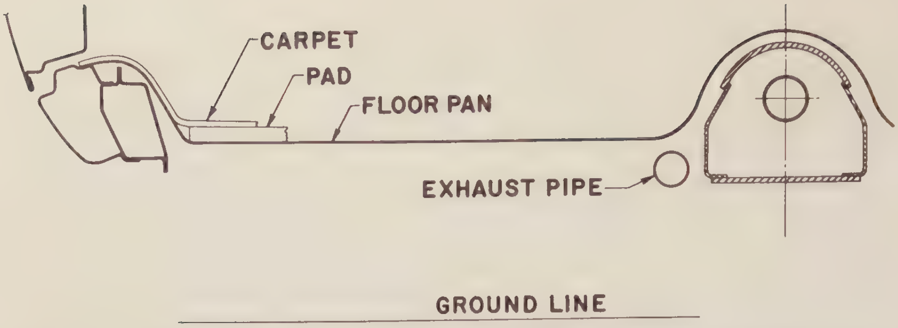


Fig. 5—This cross-section through the rear floor of a 1957 Cadillac 4-door sedan illustrates how the absence of an outer frame side rail allows a relatively narrow ledge at the rocker panel sill for ease of passenger entrance and exit.

and body arrangement have been studied and have been built experimentally. They have not, however, shown sufficient advantage in weight or height to balance the problems of cost, road noise, and non-interchangeability. Studies have indicated that, with present requirements, any advantage of integral construction applies only to the smaller cars of shorter wheelbase, and to cars for which one body model is used for the major share of the production, and over a longer period of time. Integral frame and body designs have not been adaptable economically to a variety of wheelbases and body styles because of the complicated structure which is more expensive to modify compared to separate frame and body designs. Thus, Cadillac engineers, in designing a new frame for all 1957 models, continued to direct their attention to studies of separate frame construction to meet the requirements of the trend to lower car height.

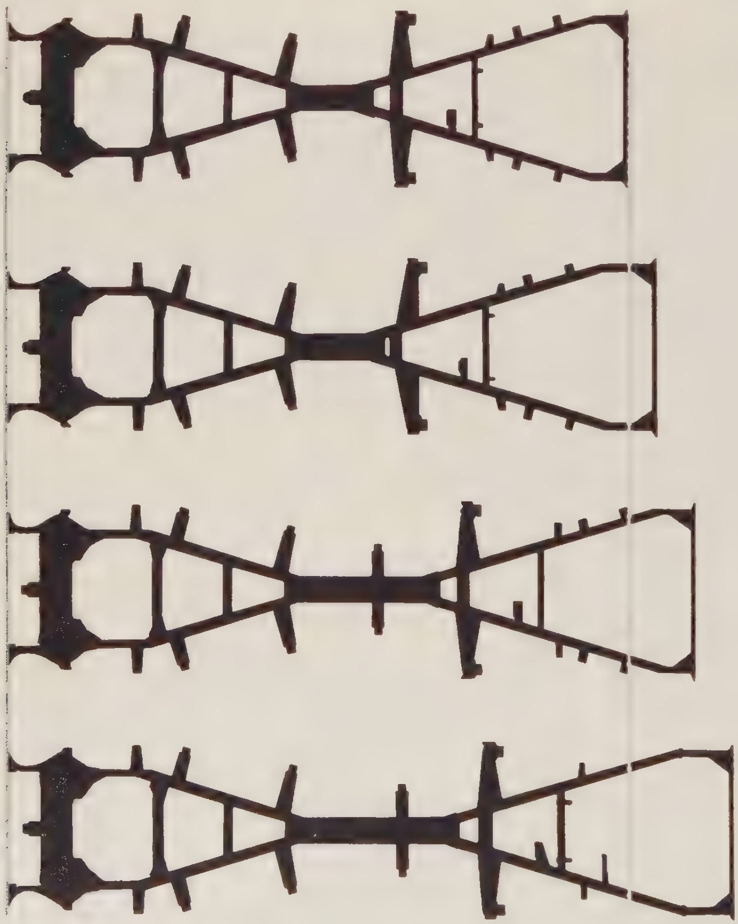


Fig. 6—The new X-frame design is adaptable to the full range of Cadillac wheelbases merely by lengthening the center section and lengthening the frame rails. Interchangeability of frame tools and parts throughout the models also is possible, which permits advantages in manufacturing. Shown above from top to bottom are frames for the Series 62 Cadillac, 129½ in. wheelbase; the Series 60, 133 in. wheelbase; the 8-passenger Series 75, 150 in. wheelbase; and the commercial chassis, 156 in. wheelbase. These use the same frame-forming tools for the front and rearward extending rails. Likewise, the same forming tools can be used for the tubular center sections which are uniform in cross section.

Completely New Frame Design Was the Best Solution

The objective in the design of the 1957 Cadillac was to reduce the height from 62 in. in the regular 1956 cars to 59 in. for the sedan models and to 57¾ in. for standard coupe models. In addition, a completely new model, the Eldorado Brougham, was proposed for 1957 and this was to have a height of 55½ in. In achieving this objective, engineers had to develop new solutions to some of the familiar problems of lowered car heights. For example, roof rails and heavy pillars, which contributed to the structure of earlier cars, have been gradually reduced or eliminated in the trend to wrap-around glass areas and hardtop styling. Bodies have become progressively heavier as more glass, more conveniences, and more accessories have been added to the modern automobile.

All of the changes of this nature have progressively imposed a greater portion of the load-resistant structure into the lower part of the car, while room for frame members has decreased. It seemed obvious, therefore, that only a co-ordinated design of frame and body would produce the necessary strength for these new automobiles.

One of the early design considerations was the possibility of utilizing the 1956 frame design in the proposed, lowered 1957 car. A study of the 1956 frame and floor-pan relationship, however, showed that the frame side rail and X-brace made it impossible to lower the body height without seriously reducing headroom or chair height (Fig. 2).

Thus, the requirements of maintaining headroom, maintaining chair height, and improving floor-pan design while maintaining structural efficiency with lower overall height led to the development of a new frame design (see Appendix).

True "X" Shape Was the Basic Design

A successful frame was developed through the efforts of engineers of the Cadillac Motor Car Division and the frame supplier. The resulting design was a frame without the conventional side

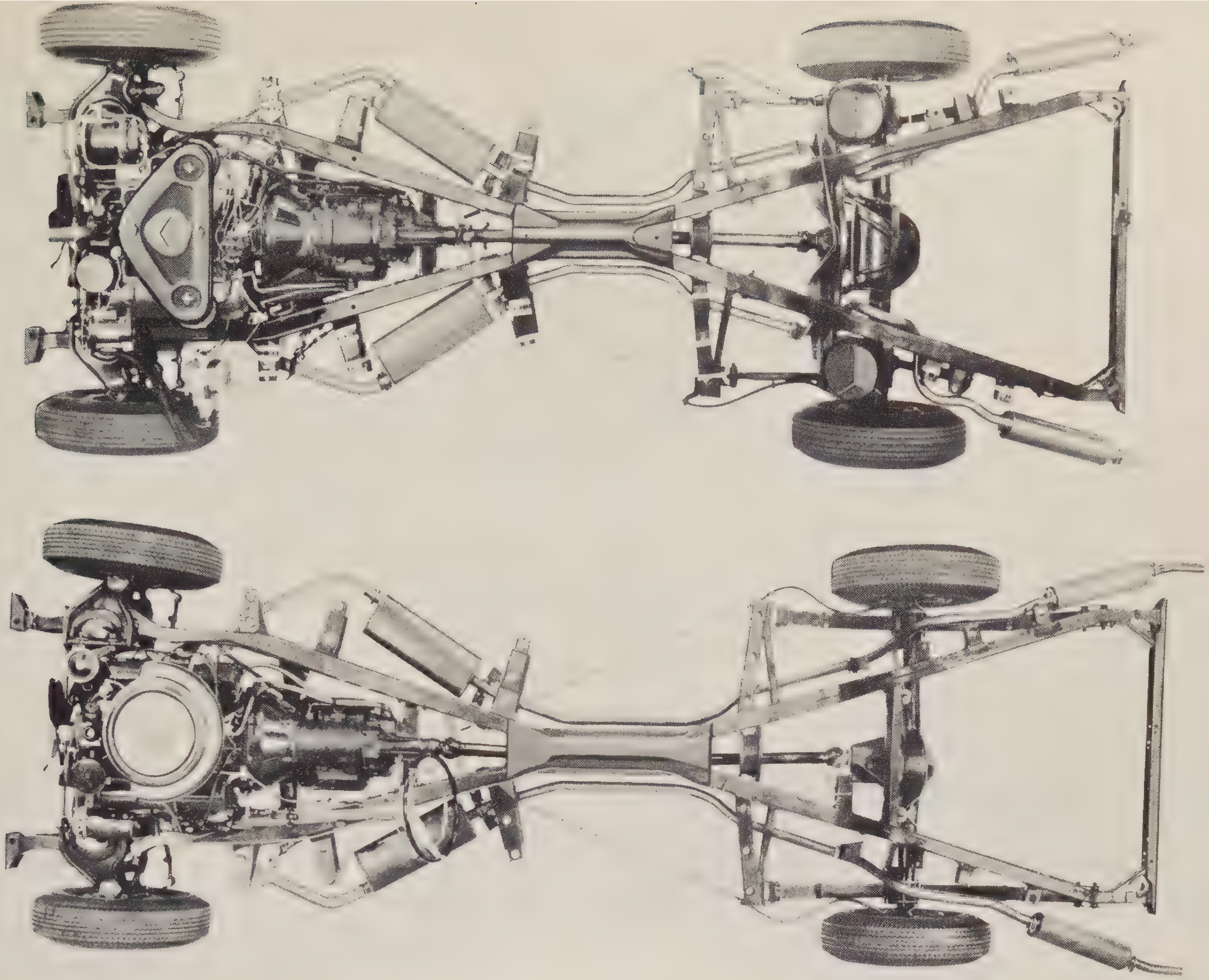


Fig. 7—The tubular center X-frame is easily adapted to use in the design of the completely different Eldorado Brougham (top), as shown in this comparison with a 1957 Series 62 Cadillac chassis (bottom). The Eldorado Brougham has air suspension on all 4 wheels, and it is shorter and lower than the Series 62. There is, however, considerable interchangeability in frame parts and mounting brackets.

rails but shaped in a true "X" using box-section members and a tubular center section (Fig. 3).

A vital part of this new frame, of course, is the intersection of the X-members. Much experimental design and testing was involved in the location and length of the tubular section at the center of the frame. Earlier designs had the junction under the front seat. This followed the practice of former frames in utilizing the height available under the front seat. But the spread of the rear, extended beams brought the frame into the floor-pan well, as occurred on the former frame type.

Further study and experimental testing showed that the junction could be moved rearward and lengthened into a tubular section. This allowed a narrow tunnel in the rear floor pan, even in the 8-passenger models where the rear seat is 28½ in. farther from the back of the front seat than in a coupe.

In the new frame design, the tunnel over the propeller shaft is only 0.4 in. higher (in relation to the propeller shaft) than for the former "X"-type frame, and the tunnel is 3 in. wider at the base. This allows a lower floor pan and a better rear-seat chair height than in the 1956 4-door sedan, without impairment of headroom. The 1956 entrance heights from seat cushion to door-opening windcord and from floor pan to windcord also were maintained in the 1957 4-door sedan. These advantages are obtained in spite of the 3 in. reduction in overall car height.

A bulkhead at the front of the tubular section contributes to the strength of the junction. Another transverse member is just behind the tubular section and between the rearwardly extending beams.

The new tubular center section provides other important advantages. With the elimination of frame side rails and the concentration of the structure at the centerline of the car, more height and width is available at the body rocker panel (outside edge of the body) for the attachment of sturdy, floor-pan cross bars (Fig. 4). For a similar reason, the anchorage for the stub pillar on 4-door hardtop models is stronger and reduced in size. Another benefit is the ease of entering and leaving the car. The ledge at the outer edge of the floor is only 6 in. wide since there is no frame side rail to clear at the body rocker sill (Fig. 5). The tubular section of the frame also provides a solid, protected mounting for the intermediate support of the propeller shaft. This support is simple, light, and inexpensive compared to the cross-member which had been required with a ladder-type frame.

Frame Has Advantages of Adaptability and Structural Efficiency

The basic shape of the tubular center X-frame has proved attractive in the

design of 1957 models because it allows not only the lower car height and lowered floor pan as desired, but also because it can be adapted to the full range of wheelbases and body styles of Cadillac cars. Engineering studies and experimental testing proved that merely by lengthening the tubular center and the frame rails the basic design could be modified for all models (Fig. 6). Furthermore, for equal weight the new frame is better than the 1956 frame in beam stiffness and torsional rigidity even in the longer wheelbase models. For example, the Series 75 frame for an 8-passenger body weighs 3½ per cent less than the 1956 frame, yet the torsional rigidity is improved 18 per cent and beam stiffness is up 16 per cent. The 1957 convertible frame, at 549 lb, weighs 6 lb less than the 1956 frame for the same model, yet it has equal torsional rigidity from front to rear axles. The distribution of rigidity also is improved, resulting in more strength in the body area from dash to rear axle. The maximum beaming deflection of the convertible frame is 0.058 in. which is 100 per cent stiffer than the 1956 model.

Several manufacturing and assembly advantages are additional results of the basic X-shape. Since the shape of the tubular center sections are the same, except for length, in all Cadillac car models, considerable interchangeability of frame tools and parts is possible.

New Eldorado Brougham Also Uses X-Frame

The Cadillac Eldorado Brougham (Fig. 1) is an entirely different model which was designed concurrently with the new 1957 models of the conventional Cadillac series. This model has a wheelbase of 126 in., which is 3½ in. shorter than that of the Series 62 Cadillac, shortest of the conventional series. The frame is ¾ in. lower than that of the Series 62. The overall car height is 55½ in., 3½ in. lower than the Series 62. The body is a true 4-door pillarless design with the rear doors hinged from the rear. The body style and all sheet metal parts are different. Air suspension is used, front and rear, rather than coil and leaf springs as on the other Cadillac models.

The desirability of building this model without special production facilities is, of course, a constant consideration. With the proper change in frame length, some changes at the rear to accommodate the air suspension and trunk compartment

*Appendix

Some Design Fundamentals of the Tubular X-Frame

A backbone structural arrangement in automobiles—as represented by the tubular center X-frame design—is not, of itself, a new thought. It has been featured in some European cars and has been given consideration in the United States. The adaptation of this structure to the design and manufacturing problems of today's American cars, however, is new.

*This Journal paper, beginning on page 28, is based on information contained in the author's paper presented at the 1957 Annual Meeting of the Society of Automotive Engineers. Part II of this S.A.E. paper described the early developmental work on the frame and was presented by J. R. Parker of A. O. Smith Corporation, supplier of the Cadillac tubular center X-frame. Grateful acknowledgment is made to Mr. Parker for the use of portions of his information in preparing this Appendix.

Much experimental work has been carried on for a number of years to develop solutions to the continuing problems of frame design. This work has resulted in the 1957 Cadillac frame which places the side members in a long "X" arrangement, crossing the "X" at the front of the rear compartment floor. This design differs from previous conventional frame designs in 2 important ways: first, the crossing point of the "X" has moved rearward from its customary position under the front seat and, secondly, the parallel side members of the frame are eliminated entirely and the X-member itself provides their function.

Further studies have revealed that this design is structurally adequate. In fact, it permits certain improvements in rigidity and strength even though certain frame-section sizes can be reduced. Although the frame structure is known to be adequate, it is still necessary, however, to study the characteristics of the complete body and frame combination. A large share of the structural stiffness of a passenger car is contributed by the body structure and particularly by the body roof.

The accompanying illustrations (Figs. 1A through 5A) describe some of the

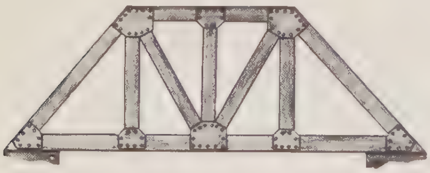


Fig. 1A—The ideal way, from a weight and stiffness standpoint, to carry bending loads is with a truss structure. Each of the members is called upon to carry only axial loads, either in tension or in compression. The large overall dimension of the assembly reduces the magnitude of these loads and results in high rigidity.

Fig. 2A—In actual practice, the application of the ideal truss structure (Fig. 1A) to the passenger automobile is rather difficult. Limitations imposed by the clearance requirements for the various parts of the vehicle, by the space and visibility requirements of the passengers, by the desire to make the car look like a thing of motion, and by the necessity for doors result in a structural compromise quite unlike the ideal, typical, modern passenger car with the bridge truss superimposed upon it. In the analogy of the truss to the body, it is apparent that a good many of the truss elements do not exist at all on the automobile body and that most of those which do exist are curved and loaded eccentrically. Each of the members must resist bending loads in varying degrees as well as direct axial loads.

floor, and some bracket and mounting changes, the new X-frame design is adapted easily to the manufacture of the Eldorado Brougham (Fig. 7).

Among the improvements which are achieved in the design of the Eldorado Brougham are the following:

- Car height is lowered 3½ in. without reducing passenger legroom, headroom, chair height and cushion size of seats
- Torsional rigidity of the Eldorado Brougham frame is 15 per cent higher and beaming stiffness is 100 per cent better than the 1956 4-door hardtop, while the frame weight increased only 19 lb—from 391 lb to 410 lb
- The overall frame and body combination has a 32 per cent higher torsional rigidity than the comparably styled hardtop of the previous year, while beaming stiffness is equal to that of a 1956 4-door sedan model. These benefits are achieved despite a 55 per cent larger windshield opening, a 50 per cent greater offset in the windshield pillar, a flatter rear-window opening, hardtop styling, and an 11 per cent heavier body
- The lower height and resulting lower center of gravity increases passenger safety because of improved car handling, particularly on turns or in cross-wind conditions.

Summary

The new Cadillac tubular center X-frame design permits a fresh concept of styling and continues the trend toward lower car bodies and their inherent safety advantages. It reverses the trend, however, toward less interior room and much heavier structures which usually are found in lower cars. The new frame design has very little penalty in weight increase, no penalty in passenger space, but provides an increase in the strength of the frame and body combination. It retains the advantages of economical manufacturing costs, design flexibility, and insulation between chassis and body, which are characteristics of separate frame construction.

Acknowledgement

The author wishes to acknowledge the contributions made by Harry M. Purdy, senior project engineer, Cadillac Motor Car Division, in the development of the tubular center X-frame.

fundamental relationships between the understructure and the superstructure of a vehicle. The illustrations relate to bending resistance only, although the frame and body also must resist torsional loads as well. The term understructure is intended to include the frame and underbody structural members, such as rocker sills and floor cross members, as well as the floor itself. The superstructure includes the remaining, principal body components, such as the roof, side frames and panels, and front end sheet metal.

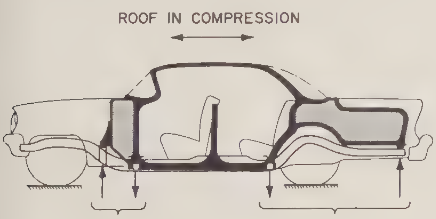


Fig. 3A—While the body roof, because of its favorable placement, does contribute importantly to the structural rigidity of the vehicle, it is far from being equivalent to the top chord of a bridge truss. In the case of the passenger car, the understructure is the principal load-carrying member, and the superstructure serves to reinforce it to the extent allowed by its own limitations and the limitations of its attachment. The reinforcing effect of the superstructure is accomplished in this manner, as shown above. The understructure, functioning as a beam, attempts to rotate in the front and rear areas. This deflection is resisted by couples in the body side-structure and by compression in the body roof. Since the roof itself is light and shallow, it offers little resistance to bending, but it can take compression to resist the bending loads on the understructure, and, similarly, it can carry shear to resist twisting loads.

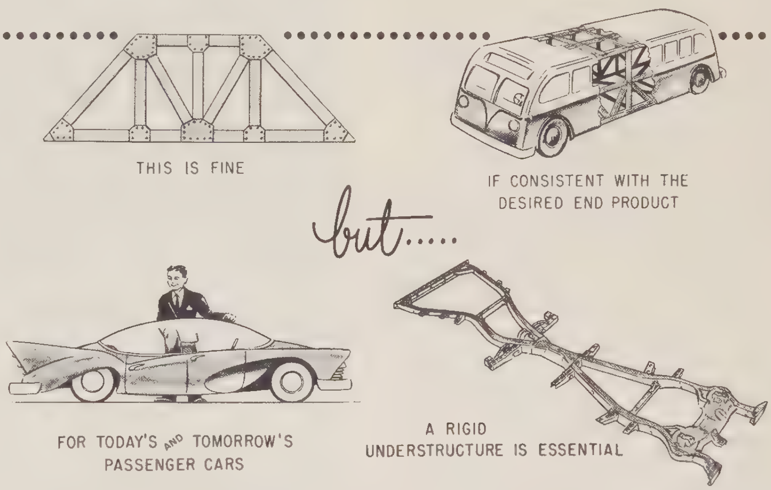


Fig. 4A—The analysis of the functioning of the various structural parts of the automobile can be resolved into 2 key points. The first of these is that the understructure functions as the principal load-carrying member, which is reinforced by the superstructure to whatever extent the capability of the superstructure will permit. This is illustrated in the above caricature. Here the familiar bridge truss can be a fair representation of the appropriate structural approach to a bus, but, in the case of a passenger car, the demands of styling and the space relationships of the various parts suggest the need of a rigid understructure.

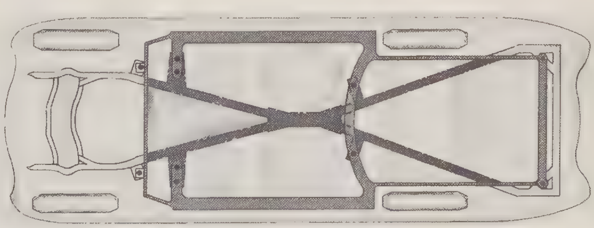


Fig. 5A—The second key point in the analysis (Fig. 4A) is that the full reinforcing effect of the body superstructure can be achieved so long as firm attachments are made at a few key locations. In order to transmit the resisting moment offered by the body roof through the side walls and inboard to the location of the tubular center X-type frame, it is necessary to have rigid cross-structure at only 4 points. Such cross-structure already exists or can be provided readily, as shown above. The body-dash serves as an admirable cross-member to translate loads from the frame location to the front of the cowl. At the front door pillar, a downward load on the body is needed, and this can be attained by a substantial outrigger bracket under the toe board with double mounting to the body. At the rear door latch pillar, another location at which the frame must load the body downwardly, a body cross-member can be provided under the rear seat, or the structure of the back of the seat can be utilized effectively. At the rear end, the frame and body structure can be brought together for suitable rigid attachments.

Application of the Analog Computer to Engineering Problems

By WALTER NOON
General Motors
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A method to simulate mistakes instead of building them

The number of electronic analog computers in this country has grown from a handful to several hundred in the past decade. The reason for this growth is simple; these computers perform a money-saving function for the user. Three and one-half years ago General Motors Process Development Staff purchased an analog computer. It has been of considerable value because of its ability to assist in the evaluation of proposed mechanical, electronic, hydraulic, and pneumatic designs. Since machines, as well as many other systems, are becoming more complex and expensive, engineers at the Process Development Staff are finding it increasingly profitable to utilize the analog computer in the solution of their problems.

Many of the problems which an engineer encounters in designing a mechanical, electrical, hydraulic, or pneumatic system can be readily solved by the application of past experiences coupled with rough calculations. However, as advances continue and more complicated systems are developed new problems are met. For the solution of many of these, experience is of limited value and rough approximations cannot be trusted. For such cases a mathematical analysis of the problem often is demanded.

Usually a skilled analyst can provide a mathematical description in the form of a set of equations which are substantially correct. However, it frequently takes more time than is available (or it may be impossible) to solve the equations and provide the engineer with quantitative data which is of use to him in actual practice. Analysis, in other words, often has been more useful as an aid in understanding a problem rather than as a means of providing a useful numerical solution.

To complete the process begun by analysis, computation is required. Until recently the main ingredient in computing was tedious and time consuming labor. In the past few years, however, tremendous inroads have been made toward reducing computing labor. High speed, multi-purpose computers now provide a means of obtaining useful, quantitative answers to complex engineering problems.

The analog computer is one of these devices. Analysts with the computer at their disposal are able not only to develop mathematical relations which

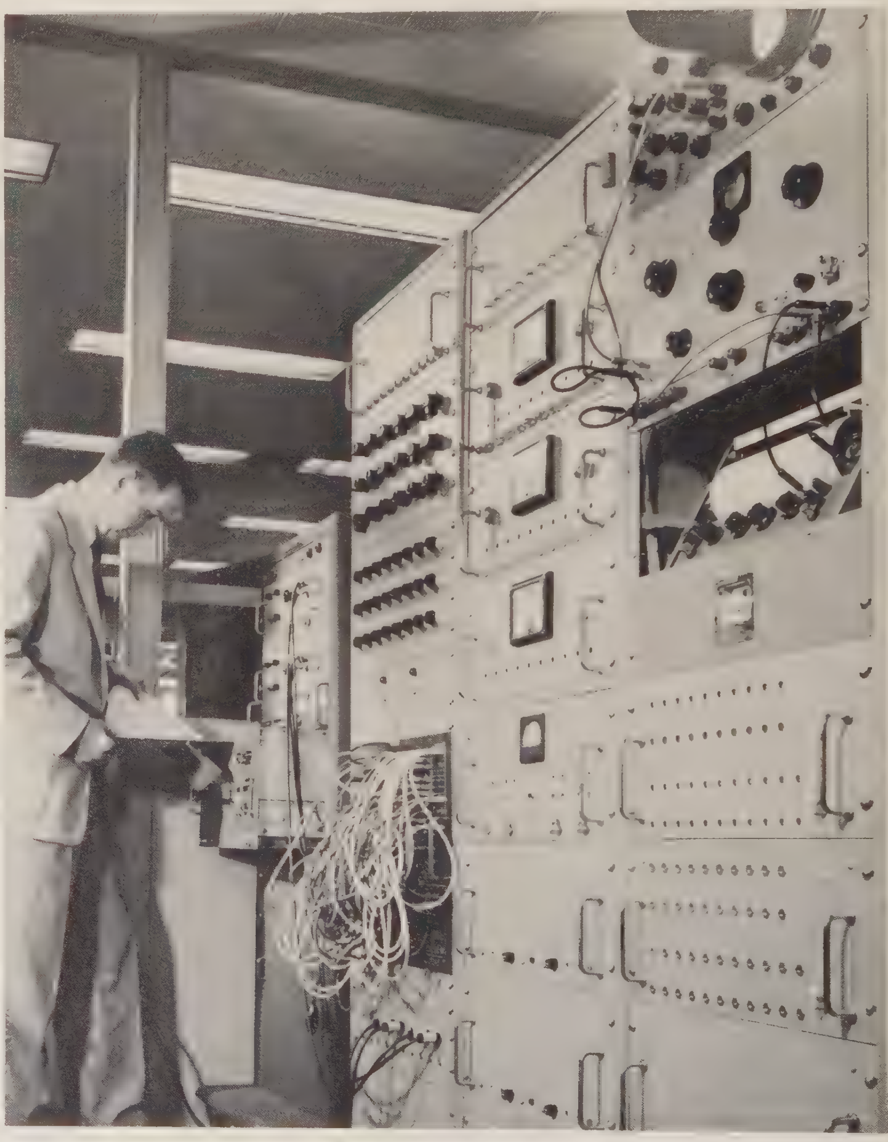


Fig. 1—This is the analog computer installation used by the Process Development Staff, GM Technical Center. The cabinet on the left is the voltage recorder. The other 3 cabinets contain the operational amplifiers, multipliers, function generators, power supplies, and associated computing equipment.

explain physical problems but also to provide specific answers in graphical form. Often, however, this is not enough. Designers sometimes ask not, "Will this system work?" but, "Which system is best?"

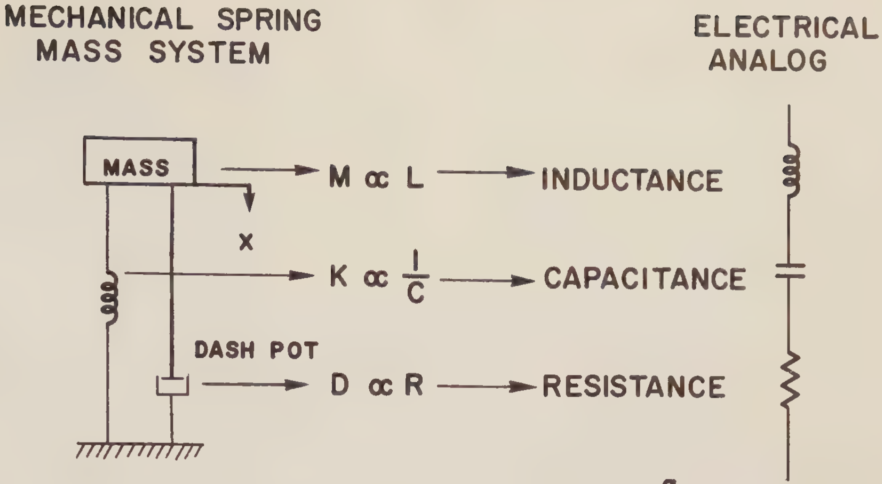
Analysis explains the general system. Computation tells how well a particular system will perform. Simulation, as provided by the analog computer, enables the analysis group to help the engineer answer the final question. The particular nature of an analog computer makes it possible to change rapidly the constants in a set of equations. Thus, in effect, components of a system are changed until the combination which gives optimum performance is found.

Briefly then, analysis, computation, and simulation can save time and money by giving precise information about a series of proposed systems before any of them are built and, therefore, reduce the trial and error time and expense involved.

Types of Analog Computers

Analog computers are widely used by government and industry. In fact, there are hundreds of analog computers now in use in this country, and the demand is increasing. These computers may be homemade, or they may be purchased in build-it-yourself kits or as complete, ready-to-use units. They vary in cost from less than $1,000 to approximately $1,000,000. The area of greatest application for the electronic analog computer has been in the aircraft industry. It is no longer practical to design modern aircraft without the use of an analog computer for simulation and design evaluation. The automotive industry also makes use of many medium size analog computer installations (Fig. 1).

The analog computer deals with physical variables, such as electrical voltages or rotating shafts. Analog computers may be divided into 2 types, direct and indirect. Direct analogy occurs when the problem variables and parameters are represented directly by machine variables and parameters. For example, there is a direct analogy that exists between a mechanical spring mass system and an electrical resistance, inductance, and capacitance circuit (Fig. 2). Indirect analog computers solve or assist in the solution of algebraic or differential equations. The slide rule is a familiar example of the mechanical indirect analog computer. This discussion is primarily concerned with another indirect analog type, the electronic analog computer.



$$M\ddot{x}+D\dot{x}+Kx = F(t) \qquad L\ddot{q}+R\dot{q}+\frac{q}{C}=E(t)$$

Fig. 2—Direct analogy exists between a mechanical spring mass system and an electrical inductance, capacitance, and resistance circuit. The quantities x, $\dot{x}$, and $\ddot{x}$ are the mass displacement, velocity, and acceleration, respectively. Similarly q, $\dot{q}$, and $\ddot{q}$ are the electrical charge, current, and rate of current charge, respectively.

The basic computing element of the electronic indirect analog computer is the operational amplifier. Using voltages as variables and appropriate electronic networks with the operational amplifier, it is possible to perform the basic mathematical operations of addition, subtraction, scale and sign change, integration and differentiation, as well as more complex mathematical operations. If function generators, curve followers, and multipliers are available also, it is possible to solve single or simultaneous linear and non-linear differential equations and algebraic equations or to perform specific mathematical operations for various purposes. There are many examples of both direct and indirect computers classified under the analog type (Fig. 3)[1].

Typical Analog Computer Problem Solution

An outline of an analog computer's study of a typical problem is a great help in clarifying the function of the computer. This outline is as follows:

(a) Presentation of the problem with all pertinent material

(b) Mathematical formulation of the problem; (several types of formulation are possible with differential equations most widely used). The engineer or researcher may need

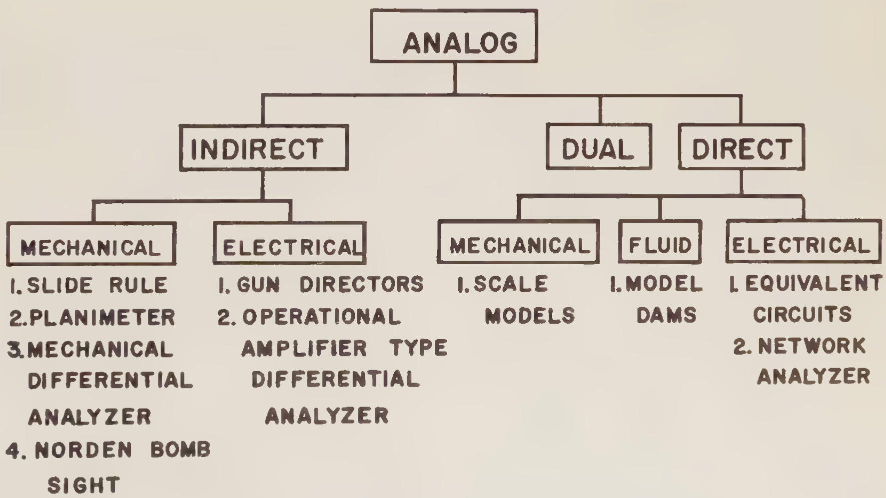


Fig. 3—Analog devices can be classified as *direct*, *indirect* or, in some cases, *dual*. The above classification is not intended to be complete; however, it indicates the variety of devices which have been developed.

the help of a mathematician or physicist in the mathematical formulation. Certain types of problems can be solved by direct simulation without mathematical formulation
(c) Preparation of the problem for analog computer solution
(d) Analog computer solution of the problem
(e) Application of results to design or analysis.

The best way to illustrate the computer's usefulness is to examine the steps necessary to obtain a computer solution to a specific problem. For example, a typical problem might concern the design of a cam.

In this problem, a cam shape was proposed to move a given load (Fig. 4). The engineer was not certain, however, that the load would be moved smoothly, and a performance evaluation was desired before the system was built. In addition, the engineer wanted to know what cam shape would be satisfactory should the proposed shape fail to provide smooth motion.

The following equation was derived by equating the forces acting on the cam follower:

$$M \frac{d^2x}{dt^2} = F_{cam} - KX - BX^3 \qquad (1)$$

where

M = effective mass of cam follower, load, and attached masses

$\frac{d^2x}{dt^2}$ = cam follower acceleration

F_{cam} = force exerted by cam on cam follower

K = spring constant of cam follower

B = non-linear spring constant of cam follower

X = cam follower displacement.

Equation (1) considers a non-linear spring. Since the cam follower displacement must be greater than or equal to the cam displacement, a further restriction on the problem was:

$$X \geqq X_{cam}$$

where

X_{cam} = cam displacement.

It is quite difficult to solve equation (1) by conventional mathematical methods, and it would be extremely time consuming to solve for a variety of cam shapes by these methods. However, this equation can be solved rapidly on the analog computer.

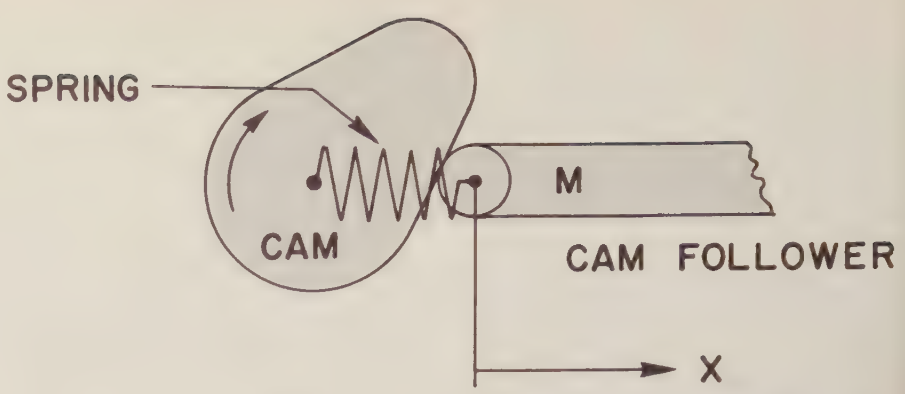


Fig. 4—Design problems in a typical cam and cam follower arrangement can be studied with the application of an analog computer. In the above example, as the cam rotates, the cam follower is displaced in the X direction only. The problem is to determine a satisfactory cam shape to give smooth motion to the transfer of the load. The computer can solve difficult equations quickly so that a satisfactory cam shape can be predicted.

For the particular problem under consideration, the cam shape was first determined by integrating the acceleration curve (Fig. 5) to obtain the velocity curve. The velocity curve was then integrated to determine the cam follower displacement. The results indicated that the proposed cam acceleration curve was unsatisfactory as the cam follower would not stay on the cam. Several other cam shapes were evaluated rapidly until a satisfactory cam shape was found (Fig. 6). If the cam designer so desired, solutions could have been obtained rapidly for various springs and masses.

Now consider the value of the computer solution to the cam design problem. The cam designer was provided with the following information:

- The proposed cam chape is unsatisfactory
- Several other cam shapes are satisfactory
- Performance characteristics of these alternate shapes are available.

This information allowed the cam designer to change the cam shape, based on predictions of actual system performance. It would have been costly and time consuming to make such changes after the system was built. This example illustrates one of the major money-saving advantages of the computer—design evaluation before construction.

Analog Computer Problem Types

An outstanding feature of the analog computer is its great versatility. Very little time is required to obtain a series of solutions for different values of problem parameters and different functions. This feature enables the engineer to explore many possible designs and "hunches" rapidly.

Accuracy is one of the fundamental considerations for an analog computer solution to a problem. The accuracy of analog computers varies from approximately 0.1 per cent to 10 per cent. Nearly all of the recent analog computers are

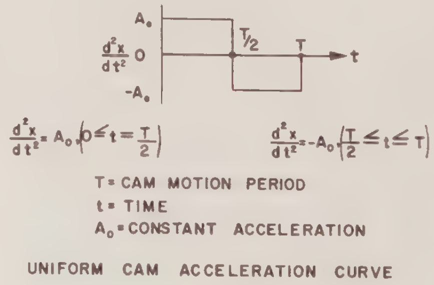


Fig. 5—This graph shows the equation solution of the cam and cam follower problem (Fig. 4). The graph indicates that the cam acceleration curve did not provide a smooth load transfer. Here, the cam follower would not stay on the cam shape proposed.

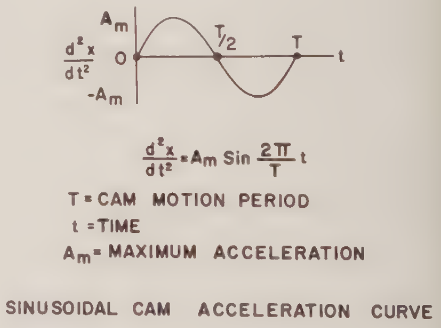


Fig. 6—To solve the cam and cam follower problem (Fig. 4), several cam shapes were evaluated by the analog computer until a satisfactory cam shape was found which would provide a smooth load transfer, as illustrated by the sinusoidal cam acceleration curve.

between 1 per cent and 0.1 per cent in accuracy. The cost of an analog computer installation increases very rapidly as accuracy is increased past one per cent[2].

An analog computer is used by the Process Development Staff at the GM Technical Center to solve a variety of problems. Most of these are dynamic problems concerned with machines—cam design, load transfer calculations, vibration analysis, heat flow calculations, and analysis of pneumatic and hydraulic systems. The solution of these problems depends on equations which accurately represent the physical system. A partial listing of the types of analog computer problems illustrates the great diversity of application (Table I).

SOME PROBLEMS SUITABLE FOR ANALOG COMPUTER STUDY
MECHANICAL AND AUTOMOTIVE ENGINEERING
Design and partial system testing of automobile control systems
Production testing of control devices
Vibrations of structures
Spring systems and vibration absorbers
Torsional vibrations in crank shafts
Dynamics of mechanisms
Hydraulic transmissions
Flow problems
ELECTRICAL ENGINEERING
Analysis of non-linear circuits
Studies of modulation systems
Studies of noise effects in communication systems
Electron optics
Feedback amplifiers and automatic control systems
Analysis of transmission lines
Servomechanism analysis
CHEMICAL ENGINEERING
Design of process controls
Continuous process analysis
Nuclear chemistry studies
RESEARCH AND ECONOMICS
Evaluation of plant efficiency
Economic distribution of power to loads from multi-station systems
Linear programming problems

Table I—This partial list of problems generally suitable for analog computer study illustrates the wide application of the computer.

Summary

Millions of dollars have been invested in electronic computers by government and industry. In return these computers perform a very valuable function for the users. This valuable function should be of great interest to engineers and research men not familiar with electronic computers. In the past, it often has been impossible to analyze and evaluate proposed designs because of the prohibitive length of time required to solve many of the equations which were developed. The arrival of electronic computers as tools to be used in the solution of these equations now makes it practical to evaluate many proposed systems, for example, flight control systems, suspension systems, servomechanisms, and assembly machines, before their actual construction.

The electronic analog computer is a powerful tool for use in the analysis of many problems because of its great versatility and relatively low operating cost. Analog computer solutions are not limited to engineering problems. The analog computer can be used in the analysis of many types of problems which have a mathematical formulation. The increasing use of computers in industry is positive proof of their effectiveness.

With the increasing demand for faster machines, better system operation, or longer life, analog computer studies are becoming essential to the design and development of many new systems. It is impossible to establish exact limits on the application of the analog computer because the primary limit is the imagination of the people concerned with the developmental projects. Applications of the analog computer can continue to expand at a rapid rate as more engineers and research men become familiar with its great potentialities.

Acknowledgement

The author wishes to acknowledge the contribution made by Kenneth E. Ruff of the GM Process Development Staff in the preparation of this paper.

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A Step in Body Manufacturing: Processing of Automotive Trim and Hardware for Production

The design of both interior and exterior appearance of all General Motors automobiles originates with the General Motors Styling Staff. Each of the 5 GM car manufacturing Divisions approves the design for its own product. The task of manufacturing the car body, with all of its styles and trim combinations to satisfy this design, falls upon the Fisher Body Division. Occupying an important position in the steps between original design and manufacture is a Fisher Body Department known as the Trim and Hardware Styling Department. While other parts of the Fisher Body organization are engaged in the structural and mechanical design of the body, the Trim and Hardware Styling Department is concerned with all items related to interior and exterior trim, such as moldings, hardware, fabrics, and paint and plastic finishes. This Department examines the original trim and hardware designs from the standpoints of cost and practicability of manufacture and prepares detailed design drawings and specifications in the proper form for use by other Fisher Body engineering and manufacturing departments. Throughout the entire procedure runs the thread of coordination—with the original designer (GM Styling), the car manufacturer (each GM car Division), and the body manufacturer (Fisher Body and suppliers).

Customer appeal is a prime factor in the first conceptions of automotive design. Its objective is to produce the most pleasing and beautiful product possible. While considering the required customer appeal, it also is necessary to keep within a stipulated price or cost range.

Fisher Body Division produces automobile bodies completely trimmed and painted for all 5 GM car Divisions: Chevrolet, Pontiac, Oldsmobile, Buick, and Cadillac. More than 75 different body styles are produced each year with 450 interior, soft trim combinations and a large number of exterior paint combinations. The General Motors Styling Staff originates the interior and exterior appearance design of all General Motors cars. After approval by the car Divisions, the body portion of the new model is forwarded to Fisher Body in the form of a drawing. While the Fisher Body Product Engineering Department is designing the structural and mechanical parts of the body, another department known as the Trim and Hardware Styling Department is concerned with the detail design and appearance of each finished part primarily from the standpoint of the practicability of manufacture.

Trim and Hardware Styling Department Organized by Specialties

The work of the Trim and Hardware Styling Department takes place over a period of time between the original conception and the finalization of design, resulting in working drawings, models, and written information. To integrate designs properly and to process them in such a manner that they can be readily assimilated by other departments, the functions of the Trim and Hardware Styling Department are subdivided into groups specializing in the various phases of trim and hardware design. These are groups specializing in (a) interior trim design, such as moldings, arm rests, fabrics, and colors; (b) interior and exterior hardware, such as door handles, lamps, ash trays; (c) interior trim and paint distribution of, for example, body cloth, floor covering, and plated or painted trim; (d) clay modeling of trim and hardware items; (e) specifications for interior finishes and plastic colors; (f) specifications for exterior painting of bodies; and (g) approval of die models.

Interior Trim Design

Original interior trim designs are created by the GM Styling Staff and presented to the car Division management for approval. After a series of meetings viewing models and materials, Divisional personnel decide on the design or designs which they desire Fisher Body to process. Drawings and sketches of these designs are then forwarded to the Trim and Hardware Styling Department with a request that a cost comparison be submitted in relation to other new designs or in comparison to present production designs.

The GM Styling trim design drawings are analyzed as to their completeness relative to detail information required to enable the appropriate Fisher Body departments to formulate estimated costs from them for presentation to the car Division. In some instances, it may be necessary to revise the original design because of structural changes required in the body. An example of this could be a problem of interior door trim design.

The repositioning of a door handle may be necessary to achieve a comfortable operating position or to satisfy the mechanical requirements of a smoothly operating window mechanism. To fulfill these requirements, the trim molding may need to be moved to give proper clearance between the molding and handle. Reduction of cost may be a prime consideration, and in this hypothetical case a careful analysis is made to determine how costs can be reduced without losing the styling characteristics of the original design (Fig. 1). Proper analysis of each interior trim design may improve high volume production at reduced costs and, yet, not deviate greatly from the original concept created by the Styling Staff.

In order to furnish the Product and Tool Cost Estimating Departments with additional information, contacts are made with other Fisher Body Engineering Departments, such as Trim Engineering, Textile Engineering, Product Drafting, the Testing Laboratory, and, when required, with engineers of Ternstedt Division. Ternstedt is the major source of supply for hardware items for all GM cars. From these conferences and past experience, the GM Styling draw-

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Who solves the problem: how to combine practicability with appealing design?

ings are finalized for the first cost estimate. These drawings and written information relative to the exact trim material specifications then are submitted to the cost estimating group.

After cost estimates have been completed, they are presented to the car Division by the Fisher Body Customer Relations Section. Should the cost be in excess of the amount the car Division desires to spend, alternate designs are developed and/or compromises are made in the original design. These compromises are worked out in conjunction with Styling Staff and the car Division involved and, when consummated, are again processed in the same manner as the original design until the car Division has been satisfied with both design and cost. They then give tentative approval to Fisher Body.

At this point, after tentative approval has been received by the Trim and Hardware Styling Department, it becomes necessary, because of inevitable additional changes, to prepare accurate quarter-scale drawings which depict the finalized trim design. These drawings show the location of stitch lines, seams, types of joints, trim finishing moldings, arm rests, and all other embellishments which complete the design. Material distribution and colors also are shown.

The drawings are submitted to the car Division for final approval. After approval, the drawings are released to all pertinent departments in Fisher Body, such as Product Drafting, Trim Engineering, and the Production Engineering Activity.

Interior and Exterior Hardware

Much the same procedure is followed in the design of hardware parts, such as interior and exterior handles, lamps, ash trays, rear view mirror supports, and other hardware necessary to make the completed body assembly. Final approval of hardware items is usually obtained from wood or metal models or working samples which show exactly the appearance of the finished article. As in the case of interior trim design, there are times when it becomes necessary to revise some of the hardware items submitted by the Styling Staff (Fig. 2). A change may be necessary because of body structure upon which the part is installed, or it may be necessary to make a revision in order to improve tooling of the part. Further, the part might be structurally weak. The car Division also may authorize suggested revisions to the design from the standpoint of reducing cost.

Again by proper analysis of hardware parts, it is possible to revise a part to make it more economical and better suited to high volume production without deviating excessively from the original design.

Several items, not technically hardware parts, may need revision for various reasons. A plastic base, applied type of door arm rest, for instance, is a part of the body assembly, although not specifically a hardware item. The arm rest must be given every consideration as to design since it also may be affected by body structure and adjacent parts (Fig. 3).

The spectre of high cost is ever present, and, therefore, serious consideration is given to the use of a minimum quantity of basic materials. High-volume production does not permit faulty design. It is vitally necessary that the simplest method of trimming be chosen without sacrificing quality or design. Every consideration must be given to proper design to assure low-cost tooling with minimum use of plastic material without sacrifice of strength. In the case of arm rests, proper placement of strengthening ribs and adequate mounting provisions in the plastic base are important to prevent any distortion which may be caused by the severe use to which arm rests are usually subjected.

After drawings of various parts and other items are completed and wood or metal models of these are approved by the car Division, they are forwarded to

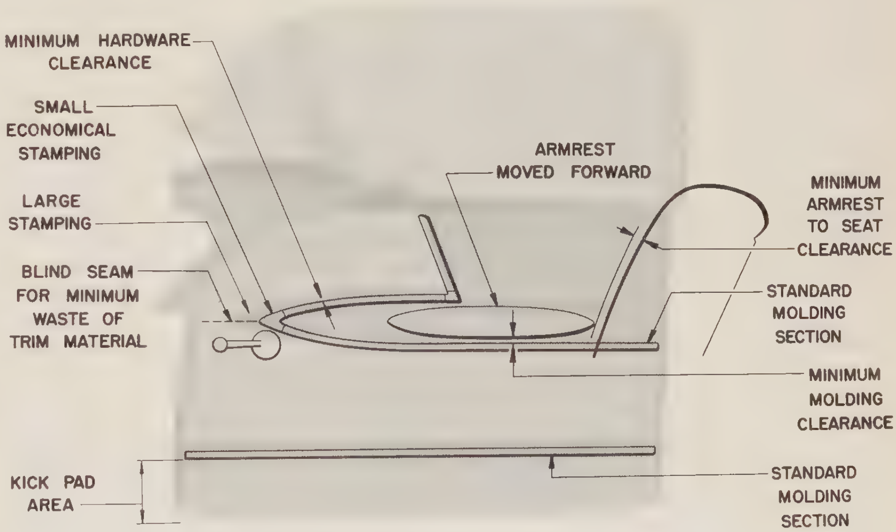


Fig. 1—A structural body change may demand an alteration of an interior design. The shaded areas depict the original design, and the line drawing illustrates the necessary revision in this hypothetical interior door trim design. The regulator handle was moved forward and up to allow the dropping glass mechanism to operate well. The trim molding then was shortened by moving it rearward and away. The arm rest was moved forward a sufficient length to achieve the proper clearance for the front seat in forward position. It will be noted that in the original design, the "spear" portion of the trim molding because of its long, tapering lines, demanded that it be manufactured as a one-piece stamping. Further analysis shows that cost can be reduced by incorporating rolled sections of constant width moldings in conjunction with a relatively small stamping at the spear end. The spear portion may overlap the rolled section molding with an exposed joint, or it can be butt-welded to the rolled section, depending on the economics involved. Costs may be reduced further by substituting existing rolled sections of smaller width in place of new, wider rolled sections as proposed for the kick pad area of the door trim pad. Another method of reducing the total cost of the trim pad is by giving consideration to the material distribution. It is apparent that the use of a horizontal stitch line, emanating from the forward point of the spear molding, reduces unnecessary waste of trim material. Without this seam it would be necessary to cut the trim material from a solid piece of stock, thereby increasing cost due to excessive use of material.

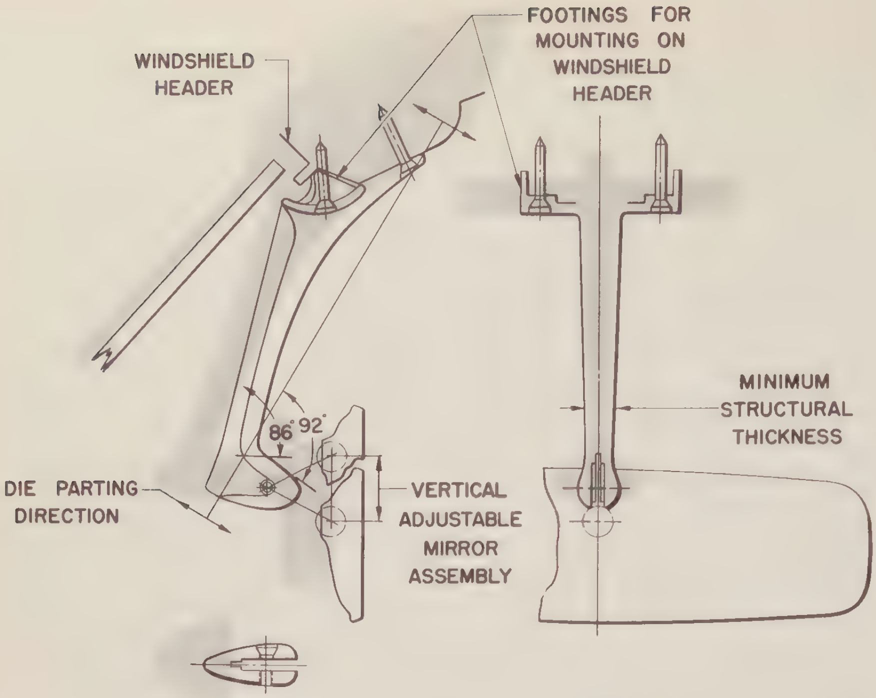


Fig. 2—Revisions to the Styling Staff original designs may be suggested by the Trim and Hardware Styling Department to improve tooling, adapt hardware items to body structure, strengthen structural weakness, or reduce piece price. The shaded areas indicate the original design, and the line drawing represents the revision of a rear view mirror support. It will be noted that, first, the *die parting angle* of 86° was changed to 92° and the wide, triangularly shaped lower end (shown in the longer view) was revised in shape to eliminate the need for a 3-part die-casting die. Increased structural strength is another result obtained from this change. The 92° parting angle permits footings for the support to be cast integrally with the arm. The footings are necessary to anchor the support arm firmly to the body structure. All this can be accomplished in a 2-part die-casting die which reduces tooling costs and saves material. Further revision may be caused by a decision on the part of the car Division to use a type of adjustable mirror which permits a greater vertical movement for driver convenience. This precludes the use of the common threaded stud and necessitates a clevis-type mirror support arm.

the Fisher Body Product Drafting Department, where production drawings are made. These, in turn, are released along with models, checking fixtures, and specifications to the Manufacturing, Tooling, and Purchasing Sections of Fisher Body Division, and to Ternstedt Division.

Interior Trim and Paint Distribution

Another function of the Trim and Hardware Styling Department is to disseminate, in package form, information relative to the total interior of the finished body. For this purpose, shaded perspective drawings of the interior to 1/10 scale are made to show the entire trim design and to show the distribution of various types of materials which make up the entire interior of the body assembly. This includes the type of trim material and area on which it is used, such as body cloth (whether plain or patterned), the direction of the cloth (if a patterned cloth), coated fabrics, genuine leather if used, floor coverings (whether carpet or rubber), and tone of materials (dark or light). In addition, the colors of interior paints, and plastics and the areas which are painted in relation to the interior trim combination used are reflected on these drawings. These drawings also show all bright molding and interior fitting, such as chrome and anodized fittings and trims.

These drawings are used for 2 major purposes. They become part of the Body Specification Manual, which contains all necessary information and references to show the car Divisions what they receive in a completed body. They also serve as a reference for use by the Quality Standards Departments in Fisher Body production plants to check completed bodies from an appearance standpoint.

Clay Modeling of Interior and Exterior Items

Another phase in designing for production is clay modeling. In order to obtain final approval on the appearance of interior and exterior moldings, full-scale half-body models and full-size models of critical areas are constructed in both wood and plaster. Three-dimensional clay moldings then are applied to show exact relationships and overall appearance characteristics. After final design approval is received, section templates and design line drawings are given to the Product Drafting Department for their use in properly showing design lines and sections on master drafts and production drawings. This requires close coordination with the various Product Drafting groups. The modeling group also makes clay, wood, and working models of hardware parts, such as ash trays, lamps, arm rests, and window stitches to determine feasibility for manufacturing. In addition, portions of the body shells are modeled in clay to prove certain aspects of design prior to the release of production drawings.

Interior Finishes and Plastic Colors

The Trim and Hardware Styling Department also issues written notices concerning all information relative to the use of interior paints, plastics, and grained or plated finishes. The original interior colors to be used on major areas are specified by the car Divisions and released to the Fisher Body Trim and Hardware Styling Department in the form of master color samples and written specifications. Working with each Division's master color panels and trim guides, Trim and Hardware Styling personnel keep the overall number of paint and plastic colors to a minimum commensurate with Divisional requirements.

Specifications for the desired type of finish and luster, such as lacquer, enamel, prime coat, plastic, chrome, and cadmium (depending on design treatment and usage) are determined and released. Bright finishes are specified in conjunction with the Testing Laboratory, while all the paint and plastic finish materials are developed in cooperation with the affected supplier and with Ternstedt Division. The Department establishes the desired master standard of finishes in order to control and maintain quality. In cooperation with Ternstedt and with outside material sources, sufficient quantities of master matching samples are developed. These are then furnished to Fisher Body production plants and related departments serving as master samples against which production parts are checked before final approval is given.

Exterior Painting of Bodies

Due to the 21 widely separated Fisher Body plant locations in which bodies are

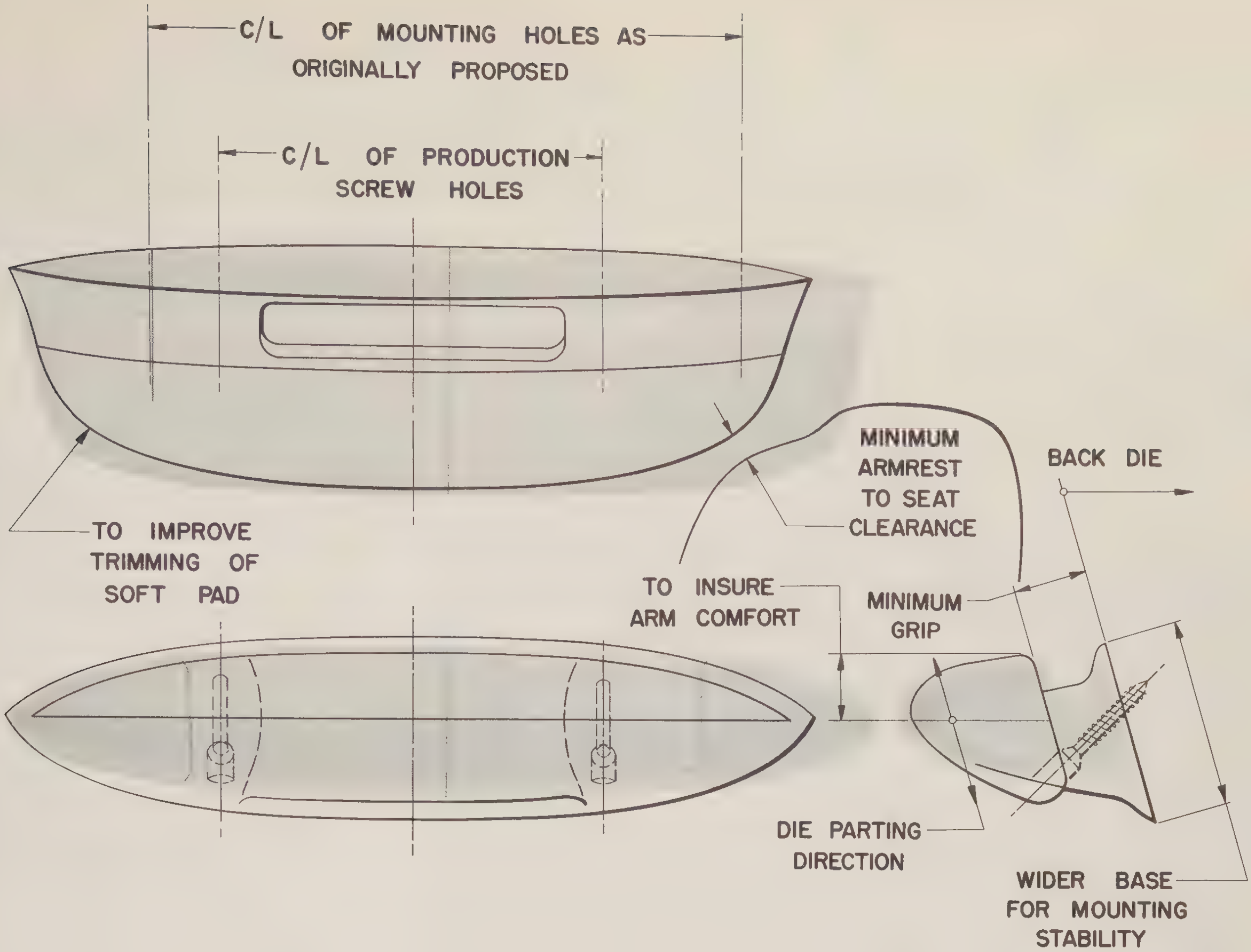


Fig. 3—A design may be revised by considering all the conditions surrounding it. For instance, in this arm rest assembly it was discovered that unavoidable interference with the front seat was caused after the seat regulator was engineered in accordance with the specification set forth by the car Division. Assuming, in this hypothetical case, that for other reasons it is impossible to move the arm rest forward to clear the front seat, it becomes obvious that the arm rest must be shortened to provide proper clearance. Other factors may be present which would bring about revision to the arm rest. It may be necessary, because of door inner panel conditions or interference with working parts within the door, to change the mounting screw holes, or it may be necessary to improve trimming conditions by "softening" the harshness of the plan view line at the extreme ends of the arm rest pad and to increase the thickness of this same pad to insure arm comfort. It also may be necessary to deviate slightly from the original design by increasing the width of the base to increase stability. Finger grip opening is important since the arm rest also serves as a pull-to when closing the door. This opening should be ample without being unsightly. The arm rest, while not specifically a hardware item, is subjected to design analysis as it may be affected by body structure and adjacent parts.

painted and the different types of bodies, the complexities of painting 2-tone and 3-tone color combinations are readily apparent. To set up standards so that bodies painted in different colors are uniform, *exterior paint color cutoff drawings* are made showing the exterior body color demarcation.

The information on these drawings has a definite effect on appearance, plant equipment, and tooling. Therefore, to satisfy all requirements before the work is started, consultations are arranged with the car Division body engineers to ascertain that the finished appearance meets with their approval. The Production Engineering Activities of Fisher Body are consulted to make certain that proper provisions are available in the production plants, and a check is made with the Paint Standards Section to determine that the requirements shown on these drawings can be obtained in a practical manner.

Approval of Die Models

To expedite drafting and facilitate tooling, the Fisher Body Engineering Section shops supply die models to the fabricators of such items as moldings and hardware parts which are irregular in shape and surface. Before shipment to the tooling sources, these models require the approval of the Trim and Hardware Styling Department to make certain that they are correct from an appearance viewpoint and reflect the original clay model.

Summary

Fisher Body serves 2 severe taskmasters—the creative stylist and the demanding customer. As the automotive stylist designs more beautiful interiors and exteriors, the body manufacturer must keep pace with improved methods of production in high volume with high quality at competitive costs.

In general, the processing of interior trim designs; the use of interior paints, plastics, and bright finishes; and the production of interior and exterior hardware must be a compromise between idealism and realism, between desire and practicability from an economic standpoint. This compromise can be met without detracting from the beauty conceived by the stylist or the interior and exterior luxury desired by the customer through an understanding and coordination of sound automotive engineering, materials available, and high-volume production techniques.

Simple Tire Mounting Machine Keeps Production Lines Moving

By ROBERT F. TUTTLE
Chevrolet Motor Division

Ingenuity solves tire mounting problem with simple, automatic machine

The scheduled production of the new 14-in. diameter Chevrolet passenger car wheel and tire required the design of a tire mounting device to accommodate this smaller size. The result was a simple machine with only 2 moving parts and fed by a continuously moving conveyor. It is automatic and requires virtually no maintenance. More importantly, it meets Chevrolet's production requirements of approximately 10 million passenger car tires per year.

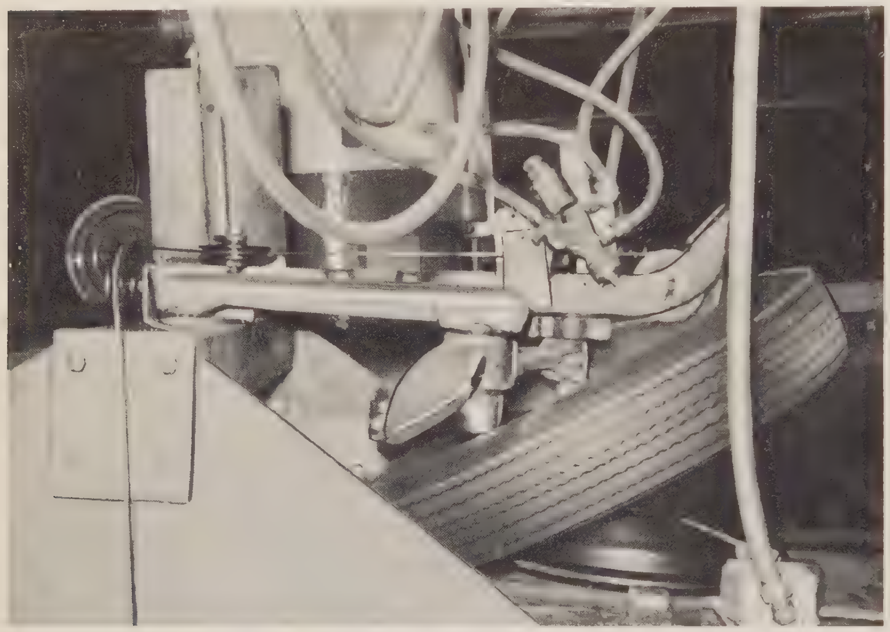


Fig. 1—This machine, formerly used to mount 15-in. diameter tire sizes, had some 60 moving parts to accomplish the assembly operation. As the wheel rims and tires progressed along on the continuously moving conveyor, they were engaged by a pair of metal paddles which threaded the beads of the tire over the rim.

When the new 14-in. diameter passenger car wheel designed with a shallow drop center was planned for production, a serious assembly problem in mounting the tire to the wheel was foreseen. Anticipating a rigorous production line schedule based on a requirement of some 10 million passenger car tires per year, Chevrolet engineers sought to devise a tire mounting machine which, in accommodating this new size, would be fast, simple, and require as little maintenance as possible.

The tire mounting machine which resulted has only 2 moving parts compared with some 60 moving parts on the machine formerly used to mount 15-in. diameter tire sizes (Fig. 1). The new machine consists of a stationary, horizontal bar located ahead of a pair of small wheels. The entire assembly is suitably mounted over a moving conveyor (Fig. 2). The machine is faster and, because of its simplicity, requires less maintenance than the old machine. In addition, it will mount both the new 14-in. and former 15-in. diameter tire sizes.

Chevrolet's new machine is located at a point in the production line just before the automobile body is dropped onto the chassis. An operator places the wheel rim outside face down over a center post on a continuously moving conveyor (Fig. 2a). The beads of the tire are lubricated before mounting, insuring that the tire will slip easily over the rim and not be cut in the process. The tire then is placed at approximately a 30° angle to the wheel with the inner and outer beads between the wheel rim (Fig. 2b).

As the tire and wheel move under the stationary cross bar, a pair of 8-in. diameter rubber wheels set parallel to the centerline of the conveyor contact the tire (Fig. 2c). These 2 wheels, which are the only moving parts of the mounting machine, serve a dual purpose: as the conveyor pulls the tire and wheel along, they squeeze and drag the leading beads of the tire back into the drop center of the rim, and, by putting the beads into the drop center, they thread the remaining portion of the tire over the rim more easily and with less stretch of the tire beads.

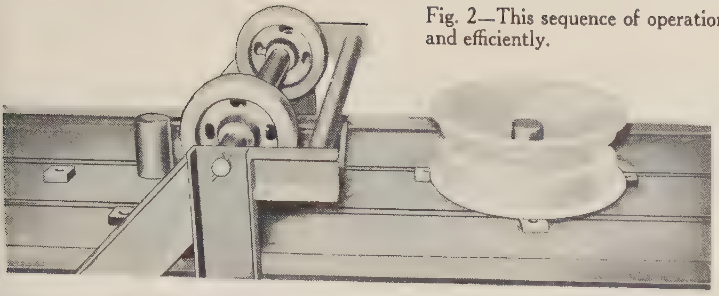
After the leading beads of the tire have been pulled into the drop center, the cross mounting bar contacts the remaining portion of the beads and threads these beads over the rim until the tire is mounted (Fig. 2d and 2e). The cross bar is set about ⅛ in. above the rim of the wheel as it moves under the bar.

As the mounted tire leaves the machine after inflation, the operator removes it from the conveyor. The mounted tire is then ready to be assembled to the car.

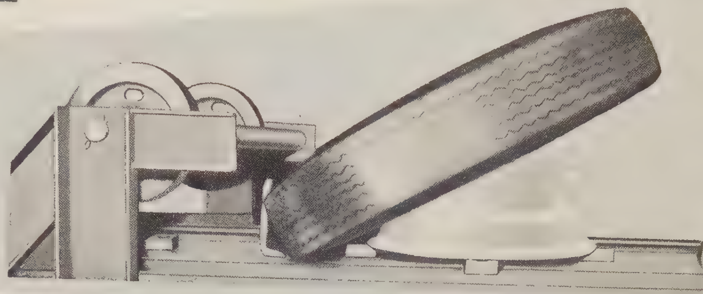
The machine is protected from possible damage by a limit switch positioned a short distance from the mounting operation. If any foreign matter should enter under the wheel so as to raise it to a height sufficient to cause contact with the cross mounting bar, the limit switch detects this difference in height and stops the conveyor before the wheel reaches the machine.

Scheduling of the machine for various colors of wheels and standard black or white wall tires is handled by the opera-

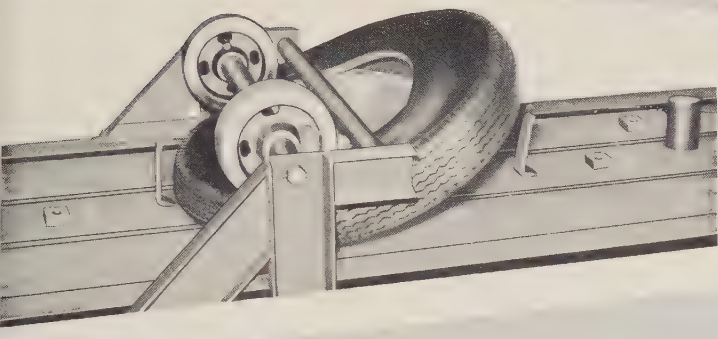
Fig. 2—This sequence of operation shows how the new tire mounting machine mounts 14-in. tire sizes simply and efficiently.



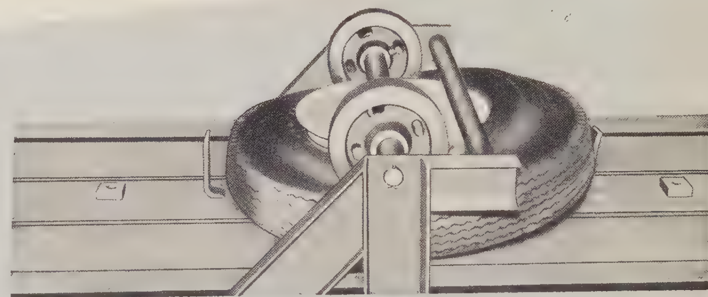
(a)—The new tire mounting machine developed by Chevrolet engineers has only 2 moving parts to mount either 15-in. diameter tire sizes or the new 14-in. sizes on drop center rims. Here the wheel rim has been placed outside face down over a center post of the continuously moving conveyor.



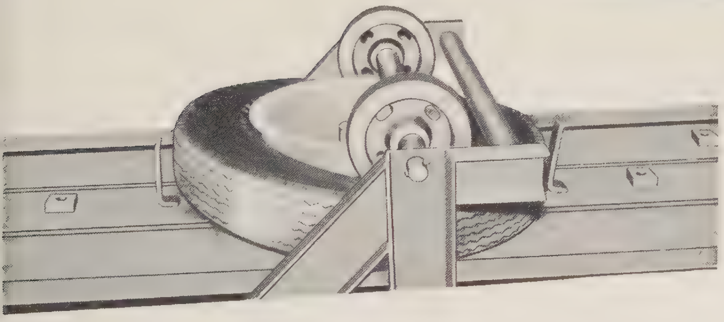
(b)—The beads of the tire are lubricated to insure that they will slip easily over the rim. The tire is set on the wheel rim with the inner and outer beads between the rim. The tire rests at an angle of approximately 30° to the rim as it approaches the stationary cross bar.



(c)—As the tire moves under the cross bar, it is engaged by 2 wheels located parallel to the centerline of the conveyor. These wheels, 8 in. in diameter, are the only moving parts of the new machine and serve to squeeze and drag the leading beads of the tire back into the drop center of the wheel.



(d)—After the leading beads of the tire have been pulled into the drop center, the cross mounting bar, which is approximately ⅛ in. above the rim of the wheel, contacts the remaining portion of the bead.



(e)—The beads of the tire are threaded over the rim until the tire is completely mounted. In the event any foreign matter should enter under the wheel so as to raise it to a height where it would strike the cross mounting bar, a limit switch (not shown) detects the increased height of the wheel and automatically stops the conveyor.

tor as he places the wheel and tire on the conveyor. There is no necessity for banking the mounted tires as the machine operates at a speed commensurate with the rest of the production line. Time-consuming maintenance is virtually eliminated due to the overall simplicity of the machine.

This tire mounting machine was developed for Chevrolet's passenger car line, but it has proved so successful since its introduction that a second machine will be used on Chevrolet's commercial vehicle assembly line.

Summary

Not content simply to modify the existing 15-in. diameter tire mounting machine, Chevrolet engineers applied ingenuity and design skill in developing a completely new type of machine to mount the new 14-in. diameter tires, as well as the 15-in. tires. Not only is this new machine faster, simpler, and more versatile, but it has proved to be the successful answer to the tire mounting operation and the vital problem of keeping the production line running smoothly.

Solution to the Previous Problem:

Determine the Pattern Design for a Shell Molded Part from a Product Drawing

By GEORGE A. HACH
Central Foundry Division
and ROBERT M. BURTON
General Motors Institute

Careful analysis of a product drawing improves pattern design

The governor body of the Hydra-Matic automatic transmission is but one of the high production castings produced by the shell molding process at Central Foundry Division. The universal objective is to produce a high-quality product economically. The shell molding process meets this objective by producing the smooth surface and accuracy required for the governor body casting. This is the solution to the problem presented in the January-February-March 1957 issue of the GENERAL MOTORS ENGINEERING JOURNAL. The solution presents the selected pattern and core designs along with the major factors considered.

THERE are several ways to approach and solve the pattern design problem for the governor body. The pattern design selected by Central Foundry (Fig. 1) is currently being used to produce governor body castings in production quantity at its Danville, Illinois, plant.

When 2 parting line alternatives present themselves, both of which require fill-ins (holes cast solid to be machined), the one generally chosen is that which will cast the contours which would be most difficult to machine. Fill-ins are used in the simpler machining operations, such as drilling or turning. Molding and coring are used on difficult contours to reduce the cost of production.

The ring grooves on the governor body can be turned, and the machined surfaces are specified on the product drawing. The oil channels would be difficult to machine in volume quantities. This would require a parting line in a plane perpendicular to the tube. The vertical location of the parting line is dictated by the location of the center lines of the cored hole and the flange ears.

The core could be parted in the plane indicated on the pattern design drawing (Fig. 1) or at 90° to that plane. The parting line shown was selected because it does not require draft or a parting line on the surfaces where the core "kisses" the shell. This results in fewer machine set-ups in producing both the pattern and the core box. The locator also can be placed in one half of the box and pattern. The locator indicated on the design drawing positions the core lengthwise and radially. This choice prevents setting the core upside down.

The detailed product drawing for machining the governor body indicated that conical machining locators are used in the ends of the cored hole. The core, therefore, is designed to overlap the ends of the hole which assures flash or fin-free locating surfaces and gives definite location and support for the core from the shell. The sharper and definite edge on the hole reduces machining.

To be consistent with good foundry practice the gate location must: (a) produce a casting free from blow holes and impurities, (b) break clean and not into the casting, (c) be accessible for "snagging" (grinding), (d) prevent "wash away" of fragile sections of the shell, and (e) create as little turbulence as possible. The tube end of the governor body, which contains the heavier section, is placed in the drag. Thus, the tube end receives the first iron poured, giving it a slightly longer time for solidification, and the feeder can be at a higher level than the section to be fed. Also, the narrow ribs of the shell are not subjected to the eroding action of the entire flow of hot iron.

The casting and core weights are calculated by breaking the work piece down into geometric sections and totaling the volumes of each section. The core weighs 0.073 lb and the casting 1.38 lb.

The pattern design for the governor body touches upon some of the conditions frequently encountered in the use of shell molds. Some of the many advantages in using the shell mold process are: (a) accuracy in shaping difficult contours, (b) increase in surface smoothness, (c) less resistance to the flow of molten iron through the use of finer sand which permits the casting of thin sections without a mis-run, (d) decrease in sand volume which reduces cost, even though the cost of shell mold sand is higher, (e) lightweight shell molds which reduce handling labor, and (f) fewer rejects due to dirt.

Many Factors Must Be Considered When Using the Shell Mold Process

Gating, feeding, and the use of risers present some of the most difficult problems to be overcome in the shell mold process. The smooth surface of the cavity, gating from the bottom, and pouring of deep molds involve controlling metal flow. Turbulence is undesirable; however, the metal must reach the cavity without having an excessive decline in temperature. At the same time, slag must be kept out of the casting cavity. The tapering of the sprue hole which receives the metal may serve as a choke. Other chokes in the gateways will check turbulence and catch dirt or other impurities. Wells may be used to absorb initial shock of the metal flow. Risers must be placed so as to furnish metal for shrinkage or fill-in. Several foundries pour shells in a horizontal position similar to the conventional green sand method. By making adapters, the foundryman is able to apply previous foundry experience and knowledge to the relatively new shell process.

Cores are made in much the same manner as molds, except that the resin-sand crust is formed on the inside of hot, split-metal core boxes. One method is to

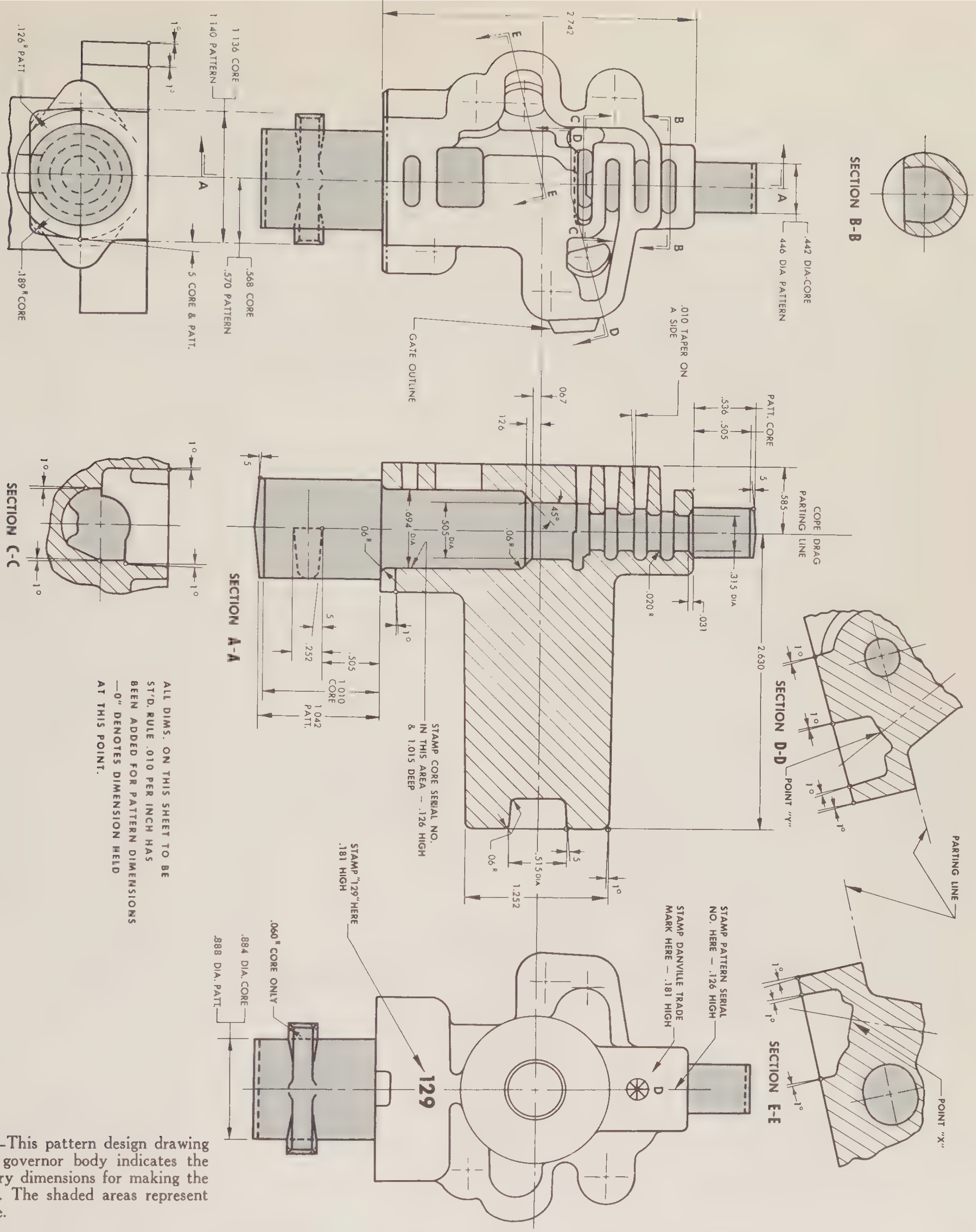


Fig. 1—This pattern design drawing of the governor body indicates the necessary dimensions for making the pattern. The shaded areas represent the core.

blow the mixture into the inverted core box with compressed air. This method allows the first part of the mixture to form a crust when it hits the heated metal box. Material not used in forming the core shell is dumped out and a hollow core results. The smaller diameter cores are solid.

Another method of core making is by dumping the sand mix into the heated core box. The core box surface must be uniformly and quickly covered to prevent imperfections in the core.

It is possible to build up composite cores as the resin-sand mixture will fuse evenly to a cured section. This process is desirable when a shell or core must be reworked.

The shell cores in shell molds are positive and dimensionally accurate. This permits accurate designing with less need for shift allowances or misalignment.

Reclamation of the sand is possible by burning out the binder. This is not practiced by all foundries using this process. However, experimentation is under way to develop other processes.

A Typical Problem in Engineering:

Determine the Design Specifications for a Pressure Regulator Coil Spring

By IVAN K. LUKEY
Buick Motor Division
and WILLIAM H. LICHTY
General Motors Institute

One spring must regulate two pressures

Controls for automatic transmissions require a variety of regulated pressures. Small, simple, balanced valves are used extensively to maintain required pressures. The regulating force can be mechanical, hydraulic, manual, or other. Where pressure regulation at a constant assigned value is needed, compact coil springs are used. The problem presented here is to determine the design specifications for a single coil spring used to control pressures of 2 different values in a torque converter automatic transmission. The spring must operate at temperatures up to 300° F and at a relatively high frequency, small amplitude oscillation.

Automatic transmissions employ hydraulic servomechanisms to operate such components as clutches, bands, and variable-angle stator blades. These devices operate at a variety of pressures. The control system must develop and accurately regulate these pressures.

The principal means of hydraulic pressure regulation used in automatic transmissions is by a relatively simple device known as a balanced valve (Fig. 1). In this type of valve, oil pressure creates a force on one end of the valve. When this force exceeds a regulating force acting on the valve at the opposite end, the valve moves against the regulating force and closes the oil-supply port. Further movement of the valve in the same direction opens an exhaust port which allows oil to escape and results in decreased oil pressure inside the valve body. The pressure decreases to a point where the regulating force again becomes great enough to move the valve in the opposite direction. This movement of the valve closes the exhaust port and opens the oil-supply port.

One example of balanced valve utilization in automatic transmissions is in regulating oil pump pressure to a specified value when the transmission is shifted to either a Direct Drive or Reverse operating position. The regulating force exerted on the valve is supplied by a single coil spring.

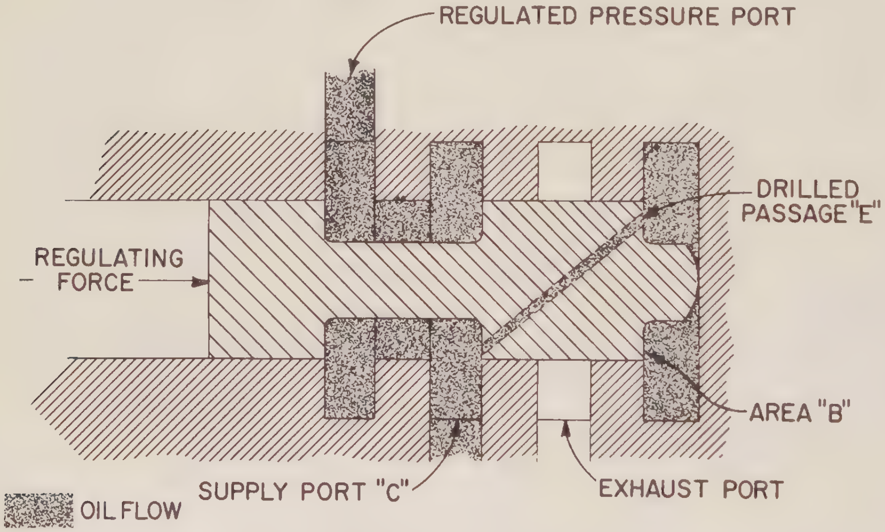


Fig. 1—A balanced valve, such as the one shown here in cross section, is the principal means used to regulate and control the operating pressures of various servomechanisms utilized in automatic transmissions. In its operation, oil at a regulated pressure enters supply port *C*, flows through the valve body, and fills the system, which requires a lower regulated pressure than that of the supply source. The oil then flows through the drilled passage *E* which results in a pressure build-up behind area *B*. When the force of this pressure, acting on area *B*, becomes greater than the regulating force acting at the left, the valve moves to the left. This movement closes supply port *C*. Further movement to the left, approximately 0.030 in. in typical designs, opens the exhaust port. Oil leakage through the exhaust port causes pressure in the output system to diminish. The regulating force again gains the upper hand and shifts the valve to the right. This movement serves to close the exhaust port and open the supply port. The overall action of the valve is such that it will oscillate with a small amplitude and a relatively high frequency, depending upon the leakage rate.

Problem

The problem is to determine the design specifications for a coil spring used as the regulating force on an automatic transmission regulator valve when operating under conditions of Direct Drive (Fig. 2a) and Reverse (Fig. 2b). The valve body design has set the following space limitations and requirements for the spring:

Maximum solid height	= 0.580 in.
Direct Drive operating length	= 0.940 in.
Reverse operating length	= 0.700 in.
Diameter of regulator valve *R* (Fig. 2)	= 0.375 in.
Maximum oil temperature	= 300° F
Maximum allowable pressure loss	= 3 psi.

The 1956 edition of the *Handbook of Mechanical Spring Design*[1] should be consulted when selecting the proper spring material to be used.

The solution to the problem will appear in the July-August-September 1957 issue of the General Motors Engineering Journal.

Bibliography

1. ......, *Handbook of Mechanical Spring Design* (Bristol, Connecticut: Associated Spring Corporation, 1956).

Other related literature in this field includes the following:

........., *Kent's Mechanical Engineers' Handbook*—Design and Production Volume (New York, New York: John Wiley and Sons, Inc., 1950, 12th Edition), Section 11.

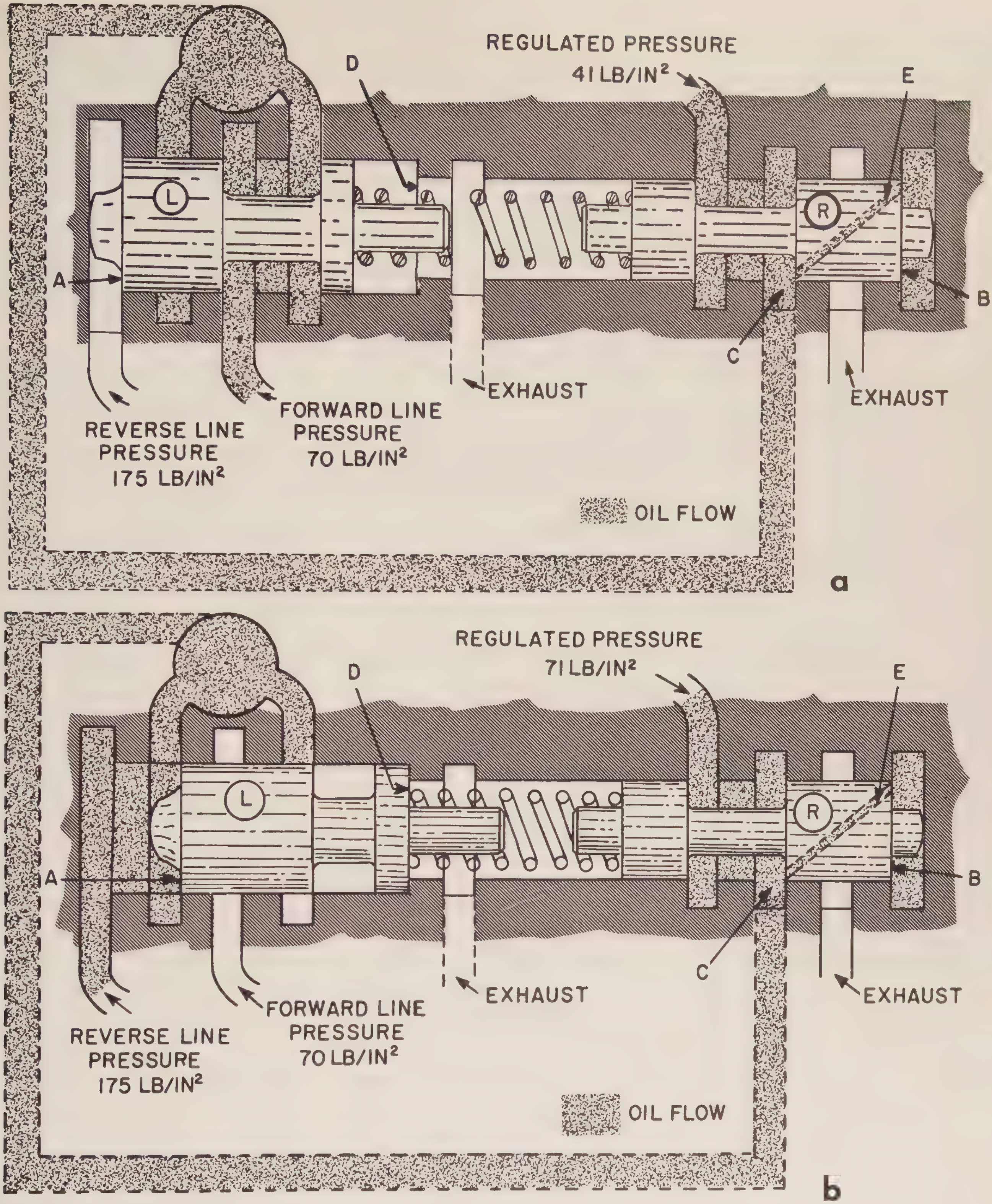
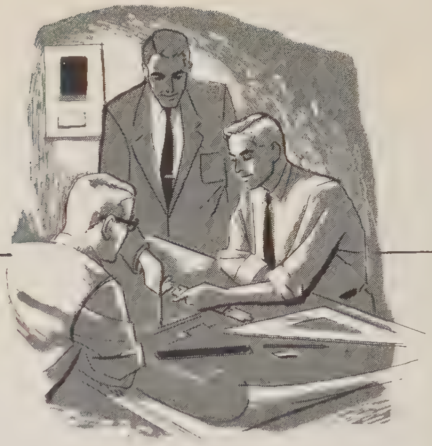


Fig. 2—One application of balanced valves in hydraulically operated automatic transmissions is in regulating oil pump pressure when the transmission is shifted to either Direct Drive (a) or Reverse (b). The right-hand valve *R* is a typical regulating valve employing a single coil spring as the regulating force. The left-hand valve *L* is a 2-position valve which performs 2 functions: it selects the pressure supply to its regulating counterpart, and it changes the length of the coil spring which acts as the regulating force on valve *R*. When the transmission is in Direct Drive operation (a), valve *L* is at the extreme left, there being no oil pressure behind area *A*. Under this operating condition, oil at a pressure of 70 psi is directed to the regulating valve *R*, where it is then stepped down and controlled at a pressure of 41 psi. When the transmission is shifted to Reverse (b), a boosted line pressure of 175 psi is allowed behind area *A*. This pressure causes valve *L* to move to the right a distance of 0.240 in. Shoulder *D* provides the stop. This movement shortens the regulator coil spring by 0.240 in. and at the same time closes the 70 psi supply line and opens a clear path for 175 psi oil to the regulating valve *R*. Under this condition, regulator valve *R* is required to limit its output pressure to 71 psi. During both operating conditions, oil enters the regulating valve *R* at the supply port *C*. The oil then flows through the drilled passage *E*, which serves to create a pressure build-up behind area *B*.

The Importance of Keeping Records of Inventions



The making of inventions to provide better living conditions is the result of painstaking efforts continued until a desired result has been achieved. Even the basic inventions which are of the greatest benefit to mankind in the production of more food, clothing, and shelter result from an evolutionary process.

In order to encourage the making of inventions, the study, ingenuity, and time required to devise and perfect a device capable of performing the desired result must be rewarded.

The making of an invention starts with the mental act of conceiving an idea of a construction capable of functioning to solve a problem with which the inventor is faced. The invention ends with the physical act of embodying that idea in an operative construction. The mental and physical phases of the act of inventing, thus, are continuous. In some instances, the conception of the idea is instantaneous. Other times it is gradual. Similarly, the physical phase, or the reduction to practice, is easy and rapid sometimes and slow and difficult in other cases. Often the steps in making an invention are separated by long periods of experimentation and disappointment. The work, however, continues until the invention is reduced to practice in an operative construction.

A patent is a Government grant given to the inventor to enable him to prevent others from using the invention for a limited period of time. It frequently happens that where there is a need for a certain improvement or function more than one party will make substantially the same invention at about the same time. In such instances the patent will issue to the party who can show that he made the invention at the earliest date.

Conception of the invention, being an act of mind, has no value as evidence until its existence has been manifested and proven by physical acts. It is possible to make a record of the tangible steps or events which show a completion of the mental concept. The various steps involved in embodying the mental concept into an operative construction, which functions to show a completion of the invention or a reduction to practice, can be recorded. These events are:

- The preparation of a sketch or drawing illustrating the device in question
- The writing of a description telling what the device is and how it functions
- The building of a model
- A description of the operation of the model.

It is important that adequate written records be made and preserved to show the dates on which the various steps involved in completing the invention are made.

In view of the fact that inventions frequently are very valuable, the uncorroborated testimony of an inventor will not be accepted in contested cases because of the danger that self-interest might induce the inventor to overstate the facts. It is important, therefore, that the invention be explained to others and that the various steps involved in proving the making of the invention be witnessed by another party who can testify to the facts relied on to show a completion of the invention.

Adequate Records Insurance Against Conflicting Claims to Invention

The early U. S. Patent Act of 1790 made no provision for determining who should receive a patent where 2 or more parties were claiming to have made the same invention at about the same time. As a result of the conflicting claims being made in about 1792 by the steamboat inventors Robert Fulton, James Ramsey, John Fitch, William Henry, William Symington, John Stevens, and others, an act was passed in 1793 providing that where 2 or more parties claimed to have made the same invention at about the same time their interfering claims for the patent would be settled by a board of 3 arbitrators. One arbitrator was to be selected by each of the parties and the other was selected by the Secretary of State.

Considerable confusion resulted from the practice of settling interferences by arbitration. Prior to 1836 patent applications were not examined for novelty. The party losing an interference proceeding through arbitration could demand that a patent be granted to him even though another patent had been granted to the winning party. Actions to repeal one of the patents could then be instituted in the Federal Courts provided such actions were filed within 3 years after the granting of the patent. In modern practice, interference proceedings to determine which of several applicants made the invention first are well defined and are conducted in the Patent Office with an ultimate appeal to the Federal Courts available to the losing party.

The need for the preparation and preservation of adequate records to show the completion of a mental act is not limited to inventions, but extends also to literature, music, and other forms of the arts which now may be protected by copyrights.

An interesting example of the confusion resulting from a failure to preserve adequate records is revealed in the writings of William Shakespeare (1564-1616), whose authorship to some 38 plays was challenged in earlier days by the proponents of Francis Bacon and, more recently, by proponents of Edward de Vere, the 17th Earl of Oxford.

Copyrights are granted now by the Government to protect an author from the unauthorized publication of his creative works. They are directed primarily to the arts, such as writings, music, paintings, sculpture, and other artistic works. Both the copyright and the patent are granted to the author and to the inventor as a public award given to encourage creative efforts and to provide publicly recognizable incentives.

By ALFRED E. WILSON
Patent Section
Central Office Staff

The time factor in patent grants: records testify to inventor's claim to invention

Contrasted with Shakespeare's failure to preserve records is the outstanding example of the preparation of adequate records in the form of note books by Leonardo da Vinci (1452-1519), one of the most versatile and gifted men of all times. His outstanding contributions to the arts in the fields of painting and sculpture and to science in the fields of architecture, hydraulics, anatomy, geology, mechanics, and botany are carefully noted in his writings and accompanying sketches.

It is interesting to note that the contributions of many of our most important inventors who did so much to decrease fear, pain, and want and to reduce drudgery were evolutionary in nature. Their dates of invention, thus, were important in enabling them to assert their claims of inventorship.

The basic essentials of food and clothing were made available more easily because of Eli Whitney's cotton gin in 1794, Cyrus McCormick's harvester in 1834, and Elias Howe's sewing machine in 1846. Yet Whitney's royalties from his cotton gin were consumed principally in unsuccessful litigation with Hodgkin Holmes, who had patented and marketed a low-priced cotton gin with a wire tooth cylinder. Whitney's unrealistic plan of leasing gins for a royalty of ⅓ of the cotton ginned by them led to commercial resistance, and, when he later reduced his royalty demands to $200 for a gin costing less than half that amount, he was unable to obtain widespread commercial acceptance of his gin. Whitney's later conception of the idea of using accurately dimensioned interchangeable parts used by him in the production of muskets for the Government kept his factory busy.

McCormick's reaper patented in 1834 was successful on the grain fields of the level prairies of the Middle West. Obed Hussey, who had developed a similar device, manufactured it for 7 years. His failure, however, to make improvements to keep pace with McCormick's improvements led to the eventual supremacy of the McCormick reaper. McCormick manufactured enough reapers to supply the market at a price which left a margin of profit over hand labor. That his operation was successful is reflected in the fact that the probate of his estate showed $10,000,000.

Howe's efforts to commercialize his sewing machine, which he had patented in 1846, were met with frustration and failure. Eventually his machine came to the attention of Isaac M. Singer, whose flashy commercializing methods resulted in a commercial boom for the device. Singer disregarded Howe and his patents, but, after 4 years of litigation which went to the Supreme Court, Howe prevailed, primarily because of the good records he had kept. Thereafter, he collected $25 per unit royalty from Singer. Howe has the distinction of being one of the few inventors whose patent was extended by special act of Congress. In a petition for a second extension near the end of the 7-year period covered by his first extension, he acknowledged having received $1,185,000 and contended that his invention was worth $150,000,000. At that time Congress considered he had been sufficiently rewarded and refused the requested extension.

Reports and Sketches, Prompt Patent Application Protect the Invention

The lesson to be learned from the struggles of earlier inventors is that inventors should not let their inventions lie dormant in their minds because, against a rival inventor who makes adequate notes, explains his inventions to others, and has the notes duly dated and witnessed, the procrastinator cannot prevail.

It is important, therefore, that inventors promptly prepare adequate sketches or drawings and descriptions of their inventions and sign and date them. These documents should be shown to competent witnesses to whom the development is explained, and the witnesses should also sign and date the documents. Diligence should be exercised in reducing the invention to practice and in filing an application if it appears to possess commercial possibilities.

Notes About Inventions and Inventors

Contributed by
Patent Section
Central Office Staff

Since June 1, 1956, the following General Motors employes have had patents granted in their names:

AC Spark Plug Division
Flint, Michigan

• **Norman J. Amlott,** (*General Motors Institute*) senior project engineer, Automotive Engineering Department, inventor in patent 2,748,891 for a cleaner silencer assembly.

• **Bertil Clason,** (*Coethen Polyteknikum, Germany, and Tekniska Skolan, Sweden*) senior project engineer, Automotive Engineering Department, inventor in patent 2,750,475 for a thermostatic switch.

• **Roy L. Bowers,** (*B.S., Michigan State University, 1930*) staff engineer, Engineering Department, and **John R. Gretzinger,** (*B.S.M.E., Purdue University, 1934*) now chief engineer, fuel systems and components, Engineering Department, Allison Division, inventors in patent 2,751,-085 for a filter for treatment of liquid.

• **John R. Gretzinger*, Joseph A. Chea,** now retired, and **Miles G. Hanson,** now retired, inventors in patent 2,754,002 for filters.

• **Clarence H. Jorgenson,** (*University of Michigan and University of Wisconsin*) administrative engineer, fuel controls and aircraft equipment, Milwaukee plant, inventor in patent 2,756,734 for a pressure regulating apparatus.

• **Ralph O. Helgeby,** (*M.E. degree, Horton School of Technology, Norway, and General Motors Institute*) staff engineer, in-

*Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.

ventor in patent 2,759,447 for a disc-type indicating instrument.

• **Karl Schwartzwalder,** (*B.Cer.E., 1930, and M.S., 1931, The Ohio State University*) director of research, Ceramic Laboratory, and **Helen B. Barlett,** (*Ph.D., The Ohio State University, 1931*) supervisor in charge of ceramic research, Ceramic Research Department, inventors in patent 2,760,875 for a ceramic composition and process for making same.

• **Wesley W. McMullen,** (*B.S.M.E., University of Michigan, 1934*) staff engineer, Automotive Engineering Department, inventor in patent 2,764,142 for an air cleaner and silencer assembly.

Allison Division
Indianapolis, Indiana

• **Floyd G. Dougherty,** (*B.S.M.E., Purdue University, 1939*) chief, aircraft gas turbine combustion group, inventor in patent 2,748,567 for a gas turbine combustion chamber with telescoping casing and liner sections.

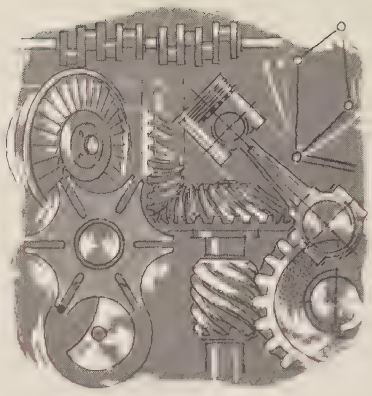
• **Dale W. Miller,** (*B.S. in education, Wittenberg College, 1934*) assistant chief engineer, Engineering Department, Aeroproducts Operations, **Calvin C. Covert,** (*B.S.M.E., University of Cincinnati, 1950*) senior engineer, Engineering Department, Aeroproducts Operations, **Darrell E. Royer,** (*Sinclair College and University of Dayton*) senior designer, Engineering Department, Aeroproducts Operations, and **Michael Demido,** (*diploma in aircraft engineering, New York State Aviation School*) designer, Engineering Drafting Department, Aeroproducts Operations, inventors in patent 2,748,877 for a propeller control with pitch lock.

• **Virgil K. Eder,** senior project engineer, Electronics and Parts Test Department, inventor in patent 2,750,817 for a drill.

• **James W. Light** (*Miami University and The Ohio State University*) experimental engineer, Engineering Department, Aeroproducts Operations, inventor in patent 2,753,031 for centrifugally responsive couplings.

• **Albert P. Dinsmore,** (*Miami University*) project and design engineer, Engineering Department, Aeroproducts Operations, inventor in patent 2,754,921 for a propeller control.

• **Kenneth L. Berninger,** (*Purdue University*) senior project engineer, Engineering Department, Aeroproducts Operations, and **William A. Weis,** (*B.M.E.,*



University of Dayton, 1938) senior designer, Engineering Department, Aeroproducts Operations, inventor in patent 2,754,922 for a variable-pitch propeller control.

• **Richard E. Moore, Harold H. Detamore,** and **Morton Brooks,** who are no longer with the Division, and **Dale W. Miller*** inventors in patent 2,756,012 for a propeller pitch stop.

• **Ross O. Michel,** (*B.S.M.E., Purdue University, 1950, and M.B.A., University of Indiana, 1951*) supervisor of tool processing, Special Process Department, Aeroproducts Operations, inventor in patent 2,756,115 for a pneumatic bearing construction.

• **Robert L. Jahnke,** (*B.Aero.E., University of Minnesota, 1939*) foreign sales engineer, Aircraft Sales Department, and **Jean R. Nelson,** not associated with the Division, inventors in patent 2,756,596 for a compressor temperature-sensing system.

• **Edgar G. Davis,** (*General Motors Institute*) manager, Bearing Department, inventor in patent 2,757,055 for a grid bearing and method of making same.

• **Arthur W. Gaubatz,** (*B.S., University of Wisconsin, 1920*) senior project engineer, Experimental Engineering Department, inventor in patents 2,757,253 for a centrifugal governor and 2,761,387 for a fuel system.

• **John B. Wheatley,** (*A.B.M.E., 1929, and M.E. in aeronautics, 1930, Stanford University*) assistant chief engineer, inventor in patent 2,759,700 for a bearing cooling system.

• **Jacob C. Schmid,** (*B.Aero.E., University of Minnesota, 1938*) assistant chief engineer, fuel systems and components, Engineering Department, inventor in patent 2,760,565 for a dual fuel system.

• **Gordon E. Holbrook,** (*B.S., Massachusetts Institute of Technology, 1939*) assistant chief engineer, Engineering Department, inventor in patent 2,761,277 for a variable-nozzle actuator.

• **Victor W. Peterson,** (*B.S.M.E., Rose Polytechnic Institute, 1939*) engineer, Advanced Design and Development Department, inventor in patent 2,761,388 for a hydraulic fluid system.

• **Richard A. Hirsch,** section head—Electra propellers, Engineering Department, Aeroproducts Operations, **Robert C. Treseder,** (*B.S.E.E., University of Utah, 1937*) assistant to the chief engineer, Aeroproducts Operations, **Harold H. Detamore*** and **Richard E. Moore*** inventors in patent 2,761,517 for a control mechanism for propellers of the contra-rotation type.

• **Robert C. Treseder*, James R. Kessler,** (*B.M.E., The Ohio State University, 1947*) section head—aircraft accessories design, Engineering Department, Aeroproducts Operations, and **Robert K. Skinner,** (*B.S.E.E., University of Cincinnati, 1943*) senior project engineer, Design Engineering Department, Aeroproducts Operations, inventors in patent 2,761,518 for a propeller pitch changing mechanism.

• **Richard A. Hirsch*** inventor in patent 2,761,519 for a propeller variable-pitch change mechanism.

• **Dale W. Miller* and Sylvan G. Hendrix,** no longer with the Division, inventors in patent 2,761,520 for a safety control for a variable-pitch propeller.

• **Charles J. McDowall,** (*B.S.M.E., University of Florida, 1927*) chief engineer, Advanced Design and Development Section, Aircraft Engineering Department, and **Oscar V. Montieth,** (*B.S.E.E., New Mexico College of Agriculture and Mechanic Arts, 1931*) chief development engineer, Production Design and Development Group, inventors in patent 2,763,462 for a turbine casing construction.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

• **George P. Fleischman,** general foreman, Transmission Model Shop, inventor in patent 2,764,260 for a vibration dampening means for brakes, clutches, and the like.

Cadillac Motor Car Division
Detroit, Michigan

• **Daniel M. Adams,** (*Detroit Institute of Technology and University of Michigan*) staff engineer, Body and Sheet Metal Section, inventor in patent 2,751,256 for a simulated wire wheel.

• **Robert W. Burton,** (*General Motors Institute*) senior project engineer, Engineering Department, inventor in patent 2,753,848 for fluid power steering.

Chevrolet Motor Division
Detroit, Michigan

• **George J. Mach,** (*Illinois Institute of Technology*) assistant staff engineer—advance truck chassis, Engineering Department, inventor in patent 2,753,947 for a forward-mounted truck cab with a movable seat.

• **Carl M. Bliss,** senior project engineer, Production Engineering Department, inventor in patent 2,760,382 for a steering column assembly.

• **John Dolza,** (*M.S. degree in E.E. and M.E., Polytechnic Institute, Turin, Italy, 1926*) now engineer-in-charge, Power Development Section, GM Engineering Staff, inventor in patent 2,760,468 for an engine cooling system.

• **Richard C. Stolte,** (*Purdue University*), motor engineer, Research and Development Department, inventor in patent 2,761,437 for an intake manifold.

• **John G. Else,** (*B.S.E.E., University of Notre Dame, 1940*) design engineer, Engineering Department, inventor in patent 2,762,352 for an automatic choke means.

• **Maurice S. Rosenberger,** (*Nebraska Western University*) assistant chief engineer in charge of experimental operations, Experimental Engineering Department, inventor in patent 2,762,384 for a governor for a hydraulically controlled automatic transmission.

Delco Appliance Division
Rochester, New York

• **Walter D. Harrison,** senior design engineer, Engineering Department, and **John B. Dyer,** deceased, inventors in patent 2,748,612 for a windshield wiper actuating mechanism.

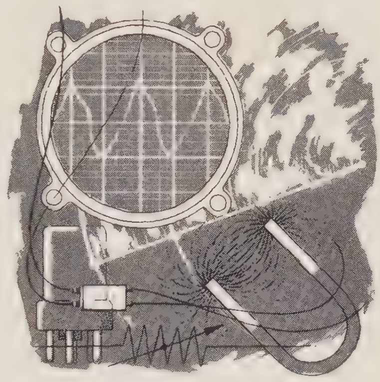
Delco Products Division
Dayton, Ohio

• **Richard F. Glock,** senior designer, Tool Design Department, inventor in patent 2,754,558 for a machine for removing molded parts from a work-piece.

• **Ralph K. Shewmon,** (*I.E. graduate, General Motors Institute, 1934*) engineering manager, Research and Development Department, inventor in patents 2,754,-579 for a method of making a motor end frame and 2,762,900 for a motor control device.

Delco Radio Division
Kokomo, Indiana

• **Bertram A. Schwarz,** technical assistant to the general manager, inventor in patent 2,751,503 for a solenoid powered tuner.



Delco-Remy Division
Anderson, Indiana

• **Argyle G. Lautzenhiser,** (*B.S.E.E., Tri-State College, 1940*) now senior project engineer, AC Spark Plug Division, **George B. Shaw,** research engineer, Engineering Research Department, and **John W. Dyer,** (*B.S.E.E., Tri-State College, 1941*) research engineer, Engineering Research Department, inventors in patent 2,749,498 for a windshield wiper and control.

• **William E. Brown,** (*General Motors Institute*) staff engineer, Product Engineering Department, and **Kenneth E. Bondurant,** (*5th year degree, General Motors Institute, 1949*) project engineer, Product Engineering Department, inventors in patent 2,750,477 for a switch.

• **Brooks H. Short,** (*B.S.E.E., 1931, and M.S.E., 1934, Purdue University*) staff engineer-in-charge, Engineering Research Department, and **George B. Shaw*** inventors in patent 2,751,464 for a temperature-responsive switch.

• **William E. Brown*, Carl O. Decker,** senior checker, Engineering Drafting Department, and **Charles W. Landes,** (*M.E. degree, International School of Correspondence, 1921*) metal working machine operator, Model Shop, inventors in patent 2,751,-468 for a switch.

• **Victor W. Yahn,** designer, Drafting Engineering Department, **J. Edward Antonidis,** (*Purdue University*) section engineer, Engineering Department, **William C. Edmundson,** (*B.S.E.E., Purdue University, 1934*) staff engineer, Product Engineering Department, and **Derother H. Dudderar,** senior designer, Drafting Engineering Department, inventors in patent 2,753,477 for an electrical brush assembly.

• **Donald G. Mahoney,** (*B.S.M.E., Purdue University, 1933*) process engineer, Delco Battery Process Department, inventor in patent 2,755,192 for a mold coat.

• **Karl P. Shetterly,** assistant to welding engineer, Process Department, and **J. Raymond Wirt,** (*B.S.E.E., Purdue University, 1930*) welding engineer, Process Department, inventors in patent 2,755,-368 for a welding method.

• **J. B. Harrison,** (*B.S.Ch.E., Purdue University, 1934*) battery plant manager, and **Garth A. Rowls,** (*B.S.M.E.*, Purdue University, 1932) product engineer—storage batteries, Battery Engineering Department, inventors in patent 2,756,269 for a battery grid.

• **J. H. Bolles,** (*B.S. in physics, Ohio Wesleyan University, 1920*) director of sales and engineering, **Charles E. Buck,** (*B.S.E.E., Purdue University, 1935*) engineer—solenoids, Engineering Department, **Brooks H. Short*,** and **Arthur G. Lautzenhiser*** inventors in patent 2,757,615 for a window washer pump.

• **George B. Shaw* and Brooks H. Short*** inventors in patent 2,759,066 for a temperature-responsive switch.

• **William C. Edmundson*** inventor in patent 2,760,094 for a dynamo electrical machine.

• **Brooks H. Short* and J. Howard Flatt,** project engineer, Engineering Department, inventors in patent 2,761,537 for a windshield wiper actuating mechanism.

• **Hilton J. McKee,** (*B.S., Ball State Teachers College, 1932*) engineer, Process Department, **Loris H. Conrad,** (*General Motors Institute*) now director, Production

Engineering Department, Rochester Products Division, **James F. Salatin,** (*4th year diploma, General Motors Institute, 1940*) electronics and control engineer, Plant Engineering Department, and **Warren M. Rider** and **James S. Burge,** no longer with the Division, inventors in patent 2,761,559 for an assembling machine.

• **Brooks H. Short*** and **Charles E. Buck*** inventors in patent 2,762,019 for an ignition coil.

Detroit Diesel Engine Division
Detroit, Michigan

• **Roger D. Wellington,** (*B.S.M.E., University of Rochester, 1930*) director of test, Engineering Laboratory, inventor in patent 2,754,050 for a rotary blower.

• **Jack E. LaBelle,** (*B.S.Ch.E., Michigan State University, 1937*) chief metallurgist, Metallurgical Department, inventor in patent 2,756,622 for a power operated torque wrench.

• **John Dickson,** (*Royal Technical College, Glasgow, Scotland*) staff engineer in charge of forward design, inventor in patent 2,759,770 for a fuel injector.

Detroit Transmission Division
Ypsilanti, Michigan

• **J. James O'Malley,** (*Wayne State University*) assistant staff engineer, Engineering Department, inventor in patent 2,749,772 for a transmission and controls.

• **Kenneth E. Snyder,** senior project engineer, Engineering Department, inventor in patent 2,751,182 for a shift valve mechanism and controls therefor.

• **Walter B. Herndon,** (*B.S.E., State College of Washington, 1928, and M.S.E., University of Michigan, 1930*) director of engineering and sales, **Kenneth E. Snyder***, and **Frank J. Condon,** (*B.C.E., University of Detroit, 1933*) staff engineer, Product Engineering Department, inventors in patent 2,761,328 for automatic transmission controls.

• **Walter B. Herndon*** inventor in patent 2,763,162 for a transmission and fluid pressure control.

• **C. W. Myers,** executive engineer, Engineering Department, inventor in patent 2,764,004 for a vibration dampener.

Diesel Equipment Division
Grand Rapids, Michigan

• **Stuart F. Kutsche,** (*B.S.M.E., Georgia Institute of Technology, 1943*) senior project engineer, Engineering Department, **Edward Orent,** (*B.S.M.E., University of Michigan, 1944*) superintendent of Diesel and aircraft activities, Production Department, and **Charles W. Helsley, Jr.,** no longer with the Division, inventors in patent 2,750,953 for a fluid flow proportioner.

• **William J. Purchas, Jr.,** (*B.S.M.E., Detroit Institute of Technology, 1933*) now chief engineer, Bearings Department of Transmissions Operations, Allison Division, and **Edward Orent*** inventors in patent 2,751,253 for an adjustable spray nozzle.

• **William J. Purchas, Jr.*** and **J. R. Vickers,** foreman of engineering laboratory, Engineering Department, inventors in patent 2,762,654 for a fuel injection device.

Electro-Motive Division
La Grange, Illinois

• **William F. Holin,** (*M.E. degree, Konstanz, Germany*) senior project engineer, Engineering Department, inventor in patent 2,748,721 for a locomotive truck.

• **Billy E. Frier,** (*B.A. in physics, Alma College, 1941*) assistant electrical control engineer, Engineering Department, inventor in patent 2,749,497 for a dynamic braking control system.

• **J. Paul Miller,** (*Purdue University and Cornell University*) now on special assignment, Engineering Department, Allison Division, and **Loren D. Britton,** export locomotive design engineer, Engineering Department, inventors in patent 2,756,690 for a railway truck.

• **Theodore C. Masel,** now retired, and **J. Paul Miller*** inventors in patent 2,756,691 for a railway truck.

• **Earl D. Smith,** project engineer, Engineering Department, inventor in patent 2,757,294 for a Diesel-electric vehicle and centering device.

• **Rudolph C. Weide,** (*M.E. degree, Berlin Technical Institute, Berlin, Germany, 1923*) senior project engineer, Engineering Department, inventor in patent 2,758,169 for an electrical switch.

• **Loren D. Britton*** and **J. Paul Miller*** inventors in patent 2,759,431 for a cooling duct arrangement.

GM Engineering Staff
Detroit, Michigan

• **Howard K. Gandelot,** (*B.S.M.E., Carnegie Institute of Technology, 1917*) engineer-in-charge, Vehicle Safety Section, inventor in patent 2,749,478 for an automatic headlight control system.

• **John Dolza*** inventor in patents 2,760,347 for a self-containing air conditioning unit in an automobile and 2,763,251 for an induction means.

• **George M. Vanator,** (*B.S., 1939, and M.S., 1941, The Ohio State University*) assistant department head, Noise and Vibration Laboratory, GM Proving Ground Section, inventor in patent 2,760,369 for a vibration analyzer.

• **Richard C. Stolte*** inventor in patent 2,760,590 for hydraulic power steering.

• **Von D. Polhemus,** (*B.S.M.E., University of Cincinnati, 1933*) engineer-in-charge, Structure and Suspension Development Group, and **Max Ruegg,** (*M.E. degree, Swiss Polytechnic Institute, Switzerland, 1921*) assistant engineer-in-charge, Structure and Suspension Development Group, inventors in patent 2,762,445 for a leaf spring suspension for vehicles.

Euclid Division
Cleveland, Ohio

• **George E. Armington,** (*B.M.E., 1925, and M.S., 1926, The Ohio State University*) director of engineering, Engineering Department, inventor in patent 2,764,203 for a tire track with differential action.

• **Raymond Q. Armington,** (*B.I.E., The Ohio State University, 1928*) general manager, inventor in patent 2,764,207 for a tire track with elastic tire, 2,764,208 for a tire track, 2,764,209 for a tire track with side thrust lugs, and 2,764,210 for a tire track with variable pin centers.

• **Walter F. Double,** development engineer, Engineering Department, inventor in patent 2,764,212 for a tire track with driving points.

Fabricast Division
Bedford, Indiana

• **James P. Bradley,** (*B.S. in chemistry, University of Illinois, 1950*) general foreman—production, Manufacturing Department, and **Robert R. Dohrmann,** (*B.S.Met.E., Purdue University, 1943*) senior process engineer, Manufacturing Department, inventors in patent 2,752,257 for an investment molding.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

Fisher Body Division
Detroit, Michigan

• **Clarence P. McClelland,** (*University of Detroit*) senior product engineer, Design and Drafting Engineering Department, inventor in patent 2,750,221 for a latch striker assembly.

• **James H. Wernig,** general director of Fisher Engineering Activities, inventor in patent 2,761,728 for a sealing means for doors.

Frigidaire Division
Dayton, Ohio

• **Kenneth O. Sisson,** (*B.S.M.E., South Dakota State College, 1936*) senior project engineer, Appliance Engineering Department, inventor in patents 2,748,585 for a domestic appliance and 2,758,685 for an agitating and spinning mechanism.

• **Sam W. Alspaugh,** master mechanic, inventor in patent 2,748,862 for an engine starter control apparatus.

• **Orson V. Saunders,** supervisor of major product line, Household Engineering Department, inventor in patents 2,749,718 for a refrigerator having adjustable door shelves, 2,758,741 for a spaced wall cabinet structure, and 2,761,289 for a refrigerating apparatus having a shelf in the door compartment.

• **Donald F. Alexander,** (*B.S.E.E., 1923, and M.S.E.E., 1926, University of Maine*) section engineer, Engineering Department, inventor in patents 2,750,755 for a refrigerating apparatus and 2,759,110 for an electrical generating system.

• **James A. Canter,** (*B.M.E., The Ohio State University, 1936*) senior project engineer, Commercial Engineering Department, inventor in patent 2,750,756 for a refrigerating apparatus for water coolers.

• **Daniel L. Kaufman,** section head, Commercial Engineering Department, inventor in patents 2,750,760 for a refrigerating apparatus and 2,752,760 for an expansive valve with bulb control.

• **Ronal H. Whyte,** (*Sinclair College*) senior engineer—clothes dryers, Appliance Engineering Department, inventor in patent 2,750,779 for a domestic appliance.

• **James D. Olcott,** (*B.S.M.E., University of Dayton, 1939*) senior project engineer, Commercial Engineering Department, inventor in patent 2,751,145 for a refrigerating apparatus.

• **Clifford H. Wurtz,** (*B.S., University of Illinois, 1929*) supervisor of major product

line, inventor in patent 2,751,147 for a temperature-responsive switch.

• **Jesse L. Evans,** (*B.S.M.E., University of Dayton, 1943*) senior project engineer, Department 604, inventor in patent 2,751,486 for a domestic appliance.

• **Francis H. McCormick,** (*B.S.E.E., Washington State College, 1915*) assistant chief engineer, inventor in patent 2,752,-694 for a domestic appliance.

• **Richard S. Gaugler,** (*B.S.Ch.E., Purdue University, 1922*) supervisor of major product line, inventor in patent 2,752,762 for a freezing device.

• **John T. Marvin,** (*B.S., 1929, and Ch.E. degree, 1935, Case Institute of Technology*) patent attorney, Patent Section Dayton office, GM Engineering Staff, inventor in patent 2,753,023 for a windshield wiper mechanism.

• **George B. Long,** (*B.S.E.E., Purdue University, 1937*) supervisor of major products line, Research and Future Products Engineering Department, and **James M. Valentine,** not with GM, inventors in patents 2,753,432 for an electrical apparatus and 2,762,893 for an electronic oven with liquid collector.

• **George C. Pearce,** (*B.S.M.E., Stanford University, 1924*) section head, Appliance Engineering Department, inventor in patents 2,754,402, 2,754,404, and 2,757,663 for domestic appliances.

• **Millard E. Fry,** (*B.S.M.E., University of Pittsburgh, 1931*) senior product engineer, inventor in patents 2,754,403 and 2,754,405 for domestic appliances.

• **James W. Jacobs,** (*B.S.M.E., University of Dayton, 1954*) section engineer, Engineering Department, inventor in patents 2,755,362 and 2,762,888 for a refrigerating apparatus and 2,759,581 for an automatic clutch manually controlled.

• **Harold B. Wallis,** now retired, and **Ronal H. Whyte*** inventors in patent 2,755,564 for a domestic appliance.

• **Edward C. Simmons,** (*University of Dayton*) senior engineer, Household Engineering Department, inventor in patent 2,755,634 for a 2-temperature refrigerating apparatus.

• **John H. Heidorn,** (*General Motors Institute*) section engineer, Engineering Department, inventor in patent 2,756,487 for a method of forming passages in a forge-welded sheet metal structure.

• **Lloyd M. Keighley,** patent attorney, Patent Section Dayton office, GM Engineering Staff, inventor in patent 2,756,-565 for an ice tray.

• **Hal C. Johnston,** (*Wayne State University and Lawrence Institute of Technology*) supervisor of major product line, Commercial Engineering Department, inventor in patent 2,757,628 for a method of making a multiple-passage heat exchanger tube.

• **Rolf M. Smith,** (*M.E. degree, University of Minnesota, 1929*) now senior project engineer, Allison Division, inventor in patent 2,757,858 for a refrigerating apparatus.

• **Marshall C. Harrold,** (*B.S.M.E., Purdue University, 1931*) senior project engineer, Engineering Department, inventor in patent 2,758,176 for an electrical apparatus.

• **Jesse L. Evans*** and **Cecil J. Prashaw,** supervisor, electric power sales, Appliance Sales Department, inventors in patent 2,758,197 for an illuminated oven.

• **Carl A. Stickel,** (*B.M.E., The Ohio State University, 1927, and LL.B., University of Dayton, 1932*) patent attorney, Patent Section Dayton office, GM Engineering Staff, inventor in patent 2,758,684 for a combined motor and brake control.

• **Carel F. Abresch,** (*M.E. degree, 1934, and marine engineering degree, 1935, Institute of Technology, the Netherlands*) senior project engineer, Appliance Engineering Department, inventor in patent 2,759,347 for a domestic appliance.

• **Charles W. Ewing,** (*B.S.Ch.E., Ohio Northern University, 1931, and M.S.Ch.E., University of Cincinnati, 1933*) senior engi-

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

neer, Manufacturing Research Department, inventor in patent 2,759,446 for a brazing mixture.

• **Leland H. Grenell,** (*B.S., Pennsylvania State College, 1924*) supervisor, Metals Section, and **Clifford H. Wurtz*** inventors in patent 2,760,346 for a refrigerating apparatus.

GM Overseas Operations Division
New York, New York

• **Jakob A. Adloff,** (*engineering degree, Rheinische Ingenieurschule, Bingen, Germany, 1927*) head of Development Section, Engineering Department, Adam Opel A.G., Russelsheim/Main, Germany, inventor in patent 2,755,874 for a motor vehicle radiator resiliently and slidably mounted.

• **Jakob A. Adloff*** and **Adam Zimmer,** (*engineering degree, Rheinische Ingenieurschule, Bingen, Germany, 1929*) senior designer, Engineering Department, Adam Opel, A.G., Russelsheim/Main, Germany, inventors in patent 2,757,016 for a front axle for an automobile with independent wheel springing.

GMC Truck & Coach Division
Pontiac, Michigan

• **Hans O. Schjolin,** (*B.S. degree, Karlstad University, Sweden, 1920, and Polytechnical Institute, Mittweida, Germany, 1923*) new development engineer, Engineering Department, inventor in patent 2,764,269 for a hydraulically operated double-acting clutch and controls therefor.

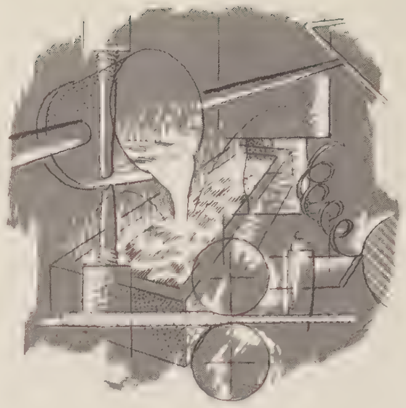
Guide Lamp Division
Anderson, Indiana

• **Robert N. Falge,** (*B.S.E.E., University of Wisconsin, 1916*) technical assistant to the general manager, inventor in patents 2,749,434 for a vehicle lamp and 2,752,481 for a sealed beam lighting unit.

• **James J. Martin,** drafting group supervisor, Product Engineering Department, and **Howard I. Slone,** (*B.S.E.E., Purdue University, 1938*) accessory engineer, Engineering Department, inventors in patent 2,752,823 for a dirigible spot lamp and mirror.

• **George W. Onksen,** (*B.I.E., General Motors Institute, 1956*) supervisor of research, and **Charles W. Miller,** (*Purdue University*) project engineer, Engineering Department, inventors in patent 2,754,452 for a headlight dimmer system.

• **Robert N. Falge*, Charles W. Miller*** and **George W. Onksen*** inventors in patents 2,760,114 for an automatic headlight dimming system time delay differential bias and 2,762,930 for a light collector.



Harrison Radiator Division
Lockport, New York

• **John H. Little,** (*B.S. in electrochemical engineering, Massachusetts Institute of Technology, 1923*) now assistant staff engineer, Engineering Department, Chevrolet Motor Division, inventor in patents 2,755,726 and 2,756,663 for heating and defrosting apparatus.

• **Harold A. Reynolds,** engineering supervisor, Radiator Section, and **Adolf Schwarz,** senior product designer, Engineering Department, inventors in patent 2,759,673 for thermostatic valves.

• **Albert B. Chapman, Jr.,** product designer, Engineering Department, **William H. Jackson,** (*B.M.E., General Motors Institute, 1949*) senior project engineer, Engineering Department, and **Robert P. McDonough,** (*B.S. in chemistry, Canisius College, 1940*) senior project engineer, Product Engineering Department, inventors in patent 2,759,709 for a combination heater and muffler.

Inland Manufacturing Division
Dayton, Ohio

• **Frederick W. Sampson,** (*M.E. degree, Cornell University, 1924*) now section engineer on special assignment, Moraine Products Division, and **Arthur J. Frei,** senior project engineer, Engineering Department, inventors in patent 2,756,566 for an ice tray.

• **Paul E. Clingman,** (*B.I.E., General Motors Institute, 1935*) supervisor, Quality and Control Department, inventor in patents 2,756,795 for a resilient locking and sealing washer, 2,758,727 for a shipping spacer, and 2,763,346 for a connector strip.

• **Thomas O. Mathues,** (*B.M.E., General Motors Institute, 1947*) chief engineer, Engineering Department, and **C. E. Nicely,** foreman, Foam Rubber Seal Department, inventors in patent 2,757,415 for a method of molding articles from foaming latex.

• **Frederick W. Sampson*** and **Robert E. Davis,** project engineer, Engineering Department, inventors in patent 2,757,520 for an ice-making apparatus.

• **John T. Marvin*** inventor in patent 2,759,575 for a connector strip.

Moraine Products Division
Dayton, Ohio

• **Roland P. Koehring,** (*Earlham College*) chief metallurgist, and **Carl E. Wilkens,** (*Sinclair College*) product engineer, Engineering Department, inventors in patent 2,748,634 for an apparatus for contour rolling.

• **Frank W. Brooks,** (*B.S.M.E., Case Institute of Technology, 1935*) project engineer, Engineering Department, inventor in patents 2,748,901 for an automatic slack adjuster for brakes and 2,762,463 for an automatic wear adjustment for brakes.

• **Frederick W. Sampson*** inventor in patents 2,749,398 and 2,749,399 for steering wheel switches and 2,757,519 for an ice-making apparatus.

• **Hooper J. Houck,** plants manager, inventor in patents 2,756,200 for a porous article impregnation and 2,762,117 for a method of forming an interlocking bushing.

• **Alfred L. Boegehold,** (*M.E. degree, Cornell University, 1915*) now consultant, GM Research Staff, **Paul J. Shipe,** (*B.S. in physical chemistry, Muskingum College, 1934, and M.S. in physical chemistry, The Ohio State University, 1938*) supervisor, Engineering Laboratory, Engineering Department, and **Athan Stosuy,** no longer with the Division, inventors in patent 2,757,446 for a method of manufacture of articles from metal powder.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

• **Holle C. Luechauer,** (*A.B., University of Cincinnati, 1927*) senior experimental chemist, Chemical, Plating, Infrared, and Emission Spectrographic Laboratories, inventor in patents 2,758,962 for a method of electroplating and an apparatus therefor and 2,761,831 for an electroplating fixture.

• **Edward J. Vosler,** master mechanic, inventor in patent 2,759,846 for a method of impregnating porous metal parts with a lower melting point metal.

• **Arthur R. Shaw,** (*B.M.E., The Ohio State University, 1937*) bearing engineer, Engineering Department, and **Robert J. Kick,** (*B.S. Met.E., Missouri School of Mines and Metallurgy, 1944*) senior project engineer, Engineering Department, inventors in patent 2,762,118 for a method for forming an interlocking bushing.

Pontiac Motor Division
Pontiac, Michigan

• **Clarence F. Smart,** (*B.S.Ch.E., University of Michigan, 1916*) metallurgist, Engineering Department, inventor in patents 2,750,333 for electrodeposition of antimony and antimony alloys, 2,751,341 for electrodeposition of lead and lead alloys, 2,755,537 for an electroplated article, and 2,764,538 for a method of plating chromium over antimony.

• **Frederick G. Torley,** (*Monmouth College*) senior checker, Engineering Department, and **William J. deBeaubien,** (*General Motors Institute, 1932*) heating, ventilating, and air conditioning engineer, Engineering Department, inventors in patent 2,751,469 for an electric switch.

GM Process Development Staff
Detroit, Michigan

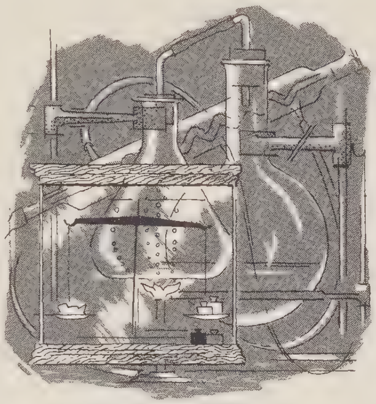
• **John M. Haldeman,** (*B.M.E., General Motors Institute*) now general superintendent, Framingham, Massachusetts, plant, B.O.P. Assembly Division, inventor of patents 2,748,453 for a method of making piston rings and 2,749,672 for a process and apparatus for grinding piston and other rings.

GM Research Staff
Detroit, Michigan

• **William S. Hackett,** (*The Ohio State University*) now research engineer, Experimental and Development Department, Central Foundry Division, inventor in patent 2,748,435 for a process for reinforcing shell molds.

• **B. John Mitchell,** (*B.S.M.E., Detroit Institute of Technology, 1940*) assistant head, Automotive Engines Department, inventor in patents 2,749,899, 2,749,900, and 2,749,901 for combustion chambers.

• **Eugene A. Hanysz,** (*B.S.E.E., 1945, and M.S.E.E., 1948, University of Michigan*) supervisor of instrument engineering, Physics and Instrumentation Department, and **Edward F. Weller, Jr.,** (*B.S.E.E., University of Cincinnati, 1943*) assistant head, Physics and Instrumentation Department, inventors in patent 2,750,791 for a thermoelectric instrument for testing materials.



• **Charles F. Froberger,** (*B.S. in chemistry, Notre Dame University 1944, and M.S. in chemistry, Michigan State University, 1950*) senior research chemist, Fuels and Lubricants Department, inventor in patent 2,751,650 for high-strength shell molds.

• **Alfred W. Schluchter,** (*B.S. in chemistry, 1919, M.S. in chemistry, 1921, and Ph.D. in physical chemistry, 1926, University of Michigan*) research engineer, Mechanical Development Department, inventor in patents 2,752,239, 2,754,202, and 2,763,546 for an aluminum-base bearing, 2,752,240 for an aluminum-base alloy bearing, and 2,756,295 for humidity sensing devices.

• **James C. Holzwarth,** (*B.S., 1945, and M.S., 1948, Purdue University*) supervisor, Metallurgical Engineering Department, inventor in patent 2,752,242 for a copper-nickel-titanium alloy and process for making same.

• **Edward J. Martin,** (*B.S., 1915, M.S. in physics, 1917, and Ph.D. in physics, 1924, University of Michigan*) head, Physics and Instrumentation Department, and Clark E. Quinn, research engineer, Physics and Instrumentation Department, inventors in patent 2,754,381 for a metal burette.

• **Alfred L. Boegehold*** inventor in patent 2,755,542 for a method of providing brazed ferrous metal parts with aluminum coating.

• **Basil D'Alleva,** (*B.S.Ch.E., University of Michigan, 1930*) research associate, Nuclear Power Engineering Department, **Albert S. Lecky,** (*Akron University*) assistant head, Technical Facilities Department, and **Victor C. Smith,** (*S.B.Ch.E., 1924, S.M.Ch.E., 1926, and Sc.D.Ch.E., 1930, Massachusetts Institute of Technology*) research associate, Nuclear Power Engineering Department, inventors in patent 2,755,663 for an engine test air supply system.

• **Paul T. Vickers,** (*B.M.E., General Motors Institute, 1948*) supervisor of Heat Transfer Section, Gas Turbines Department, inventor in patent 2,755,999 for a temperature measuring and control apparatus.

• **Robert E. Clague,** (*University of Michigan*) project mechanic, Metallurgical Engineering Department, and **Dean K. Hanink,** (*B.S.Met.E., University of Michigan, 1942*) now chief metallurgist, Metallurgical Department, Allison Division, inventors in patent 2,756,475 for an investment mold and core assembly.

• **Dean K. Hanink*** and **Robert F. Thomson,** (*B.S.M.E., 1937, M.S.M.E., 1940, and Ph.D. in M.E., 1941, University of Michigan*) head, Metallurgical Engineering Department, inventors in patent 2,757,269 for a process for brazing.

• **Charles F. Kettering,** (*B.M.E. and B.E.E., The Ohio State University, 1904*) research consultant, former general manager of the Research Staff and vice president of GM, **Kauno E. Sihvonen,** no longer with the Staff, and **Thomas C. Van Degrift,** head, Special Problems Department, inventors in patent 2,760,-023 for a humidistat.

• **Joseph B. Bidwell,** (*B.S.M.E., Brown University, 1942*) head, Engineering Me-

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chanics Department, inventor in patent 2,760,832 for a viscosity compensating system.

• **Gregory Flynn, Jr.,** (*4th year diploma, General Motors Institute, 1941*) head, Mechanical Development Department, inventor in patent 2,763,249 for an engine gas valve operating means.

• **Robert F. Thomson*** inventor in patent 2,763,519 for a powdered metal bearing.

• **Robert L. Dega,** (*B.M.E., General Motors Institute, 1948*) supervisor, Mechanical Development Department, inventor in patent 2,763,982 for a centrifugal apparatus for treating exhaust gas.

• **Charles W. Gadd,** (*B.S.M.E., Massachusetts Institute of Technology, 1937*) supervisor of vibration and stress analysis, Special Problems Department, inventor in patents 2,764,020 for a vibration pickup device and 2,764,136 for an engine oil pan.

Rochester Products Division
Rochester, New York

• **Donald D. Stoltman,** (*B.S.M.E., Rensselaer Polytechnic Institute, 1947, and M.S. in automotive engineering, Cornell University, 1948*) project engineer, Engineering Research Department, inventor in patent 2,748,760 for an engine starter control.

• **Elmer Olson,** director of sales and engineering, inventor in patent 2,749,099 for a throttle operating mechanism.

• **Clarence H. Jorgensen*** inventor in patent 2,757,258 for an automtaic circuit breaker.

• **Glenn H. Eastman,** (*Rochester Institute of Technology and University of Rochester*) sales engineer, Sales Department, **Victor H. Paine,** no longer with the Division, **Benjamin N. Snyder,** (*B.M.E., General Motors Institute, 1946*) senior project engineer, Engineering Department, and **Donald P. Worden,** process engineer-in-charge, Tubing Process Development Department, inventors in patent 2,757,732 for an apparatus for cutting tubing.

• **Lawrence C. Dermond,** (*Purdue University and Tri-State College*) senior research engineer, Engineering Department, inventor in patent 2,758,168 for an engine starting apparatus.

• **Gale F. Berninger,** supervisor of carburetor and fuel induction design, Engineering Department, **John V. Ahoe,** sales engineer, Engineering Department, and **Elmer Olson*** inventors in patent 2,762,235 for an apparatus for controlling engine idling speed.

Saginaw Steering Gear Division
Saginaw, Michigan

• **Dan R. Rowland,** (*B.S.M.E., Michigan State University, 1948*) project engineer, Engineering Department, and **Martin H. Stark,** no longer with the Division, inventors in patent 2,751,765 for a propeller shaft.

• **C. W. Lincoln,** (*B.S.M.E., University of Illinois, 1916*) chief engineer, and **Philip B. Zeigler,** (*B.S.E., Purdue University, 1941*) assistant chief engineer, Product Engineering Department, inventors in patent 2,756,605 for electric power steering.

• **David A. Galonska,** senior project engineer, and **William L. Reid,** no longer with the Division, inventors in patent 2,756,717 for a transmission control indicator.

• **Jakob A. Adloff*** inventor in patent 2,762,236 for a gear shift mechanism for transmissions.

GM Styling Staff
Detroit, Michigan

• **Robert L. Ballard,** (*B.S.M.E., 1948, and B.S.E.E., 1949, Worcester Polytechnic Institute*) now senior process engineer-in-charge of mechanization development, Process Development Department, Harrison Radiator Division, inventor in patent 2,748,687 for a moisture-responsive control means.

• **Curtis C. Whittlesey,** (*B.M.E., University of Southern California, 1944, and professional degree in industrial design, California Institute of Technology, 1948*) executive-in-charge, Fabrication and Services Department, inventor in patent 2,749,541 for an instrument panel indicator.

• **Michael J. Galla, Sr.,** assistant engineer—engineering development and drafting, Exterior Engineering Department, inventor in patent 2,757,018 for a hinged fender skirt.

• **Louis Gelfand,** (*Wayne State University*) project engineer, and **Robert F. McLean,** (*B.S.M.E., 1943, and professional degree in industrial design, 1948, California Institute of Technology*) executive in charge of styling product analysis and planning, Administration Department, inventors in patent 2,763,508 for a vehicle window regulator.

Technical Presentations by GM Engineers

Speaking appearances by General Motors engineers are one of the ways in which GM makes available to the public information on current engineering developments. Listed below are some of the recent speaking engagements by GM personnel before civic organizations, in college classrooms, and as panel members at technical society meetings. Educators who wish assistance in obtaining the services of GM engineers to speak before engineering classes or other student groups may write to Educational Relations Section, Public Relations Staff, General Motors Technical Center, P. O. Box 177, North End Station, Detroit 2, Michigan.

GM personnel who have made recent presentations are as follows:

Automotive Engineering

R. F. Sanders, assistant chief engineer, Passenger Car Chassis and Engine Department, Chevrolet Motor Division, before the Industrial Electrical Engineering Society, Detroit, Michigan, November 1; title: Why Horsepower?

K. W. Lesher, project engineer, Product Engineering Department, Diesel Equipment Division, before the Society of Automotive Testers, Mt. Vernon, Illinois, November 5; title: Design, Operation, and Servicing of Hydraulic Valve Lifters.

R. J. Wirshing, head, Chemistry Department, GM Research Staff, before the National Paint, Varnish and Lacquer Association, Los Angeles, Caifornia, November 12; title: What We Expect of Finishes for the Automotive Industry.

Robert R. Mandy, supervising engineer, Air Conditioning Section, Automotive Air Conditioning Department, Harrison Radiator Division, before the student branch of the American Society of Mechanical Engineers, Worcester Polytechnic Institute, Worcester, Massachusetts, November 13; before the Albany Chapter of the American Society of Refrigerating Engineers, Albany, New York, January 21; and before the Buffalo Chapter of the A.S.R.E., Buffalo, New

York, February 7; title: Future Trends in Automotive Air Conditioning.

G. L. Leithauser, supervisor, Chemistry Department, GM Research Staff, before the National Association of Corrosion Engineers, Detroit, Michigan, November 15; title: The Role of Finishing in the Prevention of Automotive Body Corrosion.

B. J. Mitchell, assistant head, Automotive Engines Department, GM Research Staff, before the American Chemical Society, Philadelphia, Pennsylvania, November 15; title: The Combustion Chamber, Octanes and Efficiency.

Robert K. Hathaway, service engineer, Service Department, Rochester Products Division, before the Cornell University Engineering Club, Ithaca, New York, November 27; title: Fuel Injection.

William H. Jackson, senior project engineer, Automotive Air Conditioning Department, Harrison Radiator Division, before the Chicago Section of the A.S. M.E., Chicago, Illinois, December 5; title: 1957 Automobile Air Conditioning.

David W. Eddy, senior project engineer, Spark Plug Engineering Department, AC Spark Plug Division, before the spark plug symposium, Detroit Arsenal, Detroit, Michigan, December 12; title: Discussion of Spark Plug Fouling in Ordnance Vehicles.

K. T. Kitchen, equipment sales engineer, Chicago Equipment Sales Office, AC Spark Plug Division, before the student chapters of the Society of Automotive Engineers and the A.S.M.E., University of Wisconsin, Madison, Wisconsin, December 12; title: Spark Plug Story with accompanying film.

Kenneth F. Lingg, service manager, Service Department, Rochester Products Division, before the Automotive Electric Association convention, Chicago, Illinois, December 16; title: Fuel Injection.

Theodore H. Redman, standards engineer, Product Engineering Department, Rochester Products Division, before the Brockport Lions' Club, Brockport, New York, January 7; title: Fuel Injection.

H. K. Gandelot, engineer-in-charge, Vehicle Safety Section, GM Engineering Staff, before the Central Michigan Law Enforcement Association, Lansing, Michigan, January 10; title: Engineering Advancement in the Safety of Automobiles.

Zora Arkus-Duntov, research development engineer, Chevrolet Motor Division, before the S.A.E. annual meeting, Detroit, Michigan, January 15; title: The General Motors Fuel Injection System: Development for Chevrolet Corvette Engine.

Richard C. Cook, assistant chief engineer, Engineering Department, Buick Motor Division, before the Singer Manufacturing Company Engineering Society, Elizabethport, New Jersey, January 15; title: How a Buick Is Engineered.

John Dolza, engineer-in-charge, Power Development Section, GM Engineering Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 15; title: The General Motors Fuel Injection System: Basic Development.

Ellsworth A. Kehoe, assistant chief engineer, Product Engineering Department, Rochester Products Division, before the S.A.E. annual meeting, Detroit, Michigan, January 15; title: The General Motors Fuel Injection System: Production Development.

J. F. Verkerke, senior project engineer, Pontiac Motor Division, before the S.A.E. annual meeting, Detroit, Michigan, January 15; discussion member: Fuel Injection.

O. K. Kelley, engineer-in-charge, Transmission Development Section, GM Engineering Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 16; title: The Chevrolet Turboglide Transmission.

S. L. Milliken, staff engineer, Future Design Department, Cadillac Motor Car Division, before the S.A.E. annual meeting, Detroit, Michigan, January 16; title: The Cadillac Frame—A New Design Concept for Lower Cars.

Joseph B. Bidwell, head, Engineering Mechanics Department, GM Research Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 18; group member presenting: Valve Train Wear as Affected by Metallurgy, Driving Conditions, and Lubricants.

W. R. Houser, supervisor of dynamometer, Spark Plug Engineering Department, AC Spark Plug Division, before the S.A.E. annual meeting, Detroit, Michigan, January 18; discussion of article on Spark Plug Fouling written by Ethyl Corporation personnel.

Diesel Engines

Roger D. Wellington, director of test, Engineering Laboratory, Detroit Diesel Engine Division, before the S.A.E. national Diesel engine meeting, Chicago, Illinois, November 1; title: Cyclic Testing to Simulate Diesel Engine Service Conditions.

Robert A. Pejeau, senior project engineer, Engineering Department, Cleveland Diesel Engine Division, before the S.A.E. national Diesel engine meeting, Chicago, Illinois, November 2; title: Sulfur, Snorkel, and Submarines (progress report of the Coordinating Research Council).

F. G. Rounds and **H. W. Pearsall,** senior research chemists, Fuels and Lubricants Department, GM Research Staff, before the S.A.E. national Diesel engine meeting, Chicago, Illinois, November 2; title: Diesel Exhaust Odor—Its Evaluation and Relation to Exhaust Gas Composition.

Paul J. Louzecky, mechanical section engineer, Engineering Department, Cleveland Diesel Engine Division, before the A.S.M.E. annual convention, New York, New York, November 28; title: Design and Development of a 2-Cycle Turbocharged Diesel Engine.

Arthur F. Underwood, manager, Research Staff Activities, GM Research Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 14; title: The Continental 750-hp Air-Cooled Diesel Engine.

V. C. Reddy, director of development, Engineering Department, Detroit Diesel Engine Division, before the S.A.E. annual meeting, Detroit, Michigan, January 17; title: Turbocharging the Series 71 Engine.

Foundry Methods

Elmer E. Braun, works manager, Central Foundry Division, before the American Foundrymens Society, Windsor, Canada, November 23; Canton, Ohio, January 3; and Rochester, New York, January 8; title: Opportunities of the Future; and before the Michigan regional foundry conference, Ann Arbor, Michigan, November 29; title: The Development of the Shell-Cast ArmaSteel Crankshafts.

L. J. Pedicini, staff engineer, Foundry Department, Process Development Section, GM Process Development Staff, before the Michigan regional foundry conference, Ann Arbor, Michigan; November 29; title: An Evaluation of Diaphragm Molding.

Harold G. Sieggreen, chief engineer, Central Foundry Division, before the

A.S.M.E., New York, New York, November 29; title: A New Look at Shell Castings.

R. C. Robinson, superintendent of standards and methods, Engineering Department, Central Foundry Division, befor the S.A.E., American Foundrymens Society, and A.S.M.E., Champaign, Illinois, December 12; title: The Manufacture of ArmaSteel Crankshafts.

General Engineering

Karl Schwartzwalder, director of research, Research Department, AC Spark Plug Division, before the American Ceramic Society, Clemson University, Clemson, South Carolina, November 1: title: Basic Science Division; before the A.C.S., University of Illinois, Urbana, Illinois, November 8; member: Porcelain Enamel Insititute forum; and before the Canadian Ceramic Society, Niagara Falls, Canada, February 11; title: Ceramics, Our Future.

Kenneth A. Meade, director, Educational Relations Section, GM Public Relations Staff, before the Research and Development Section of the American Drug Manufacturers Association, Edgewater Park, Mississippi, November 2; title: Opportunities in the Field of Science; and before the Foundry Club of General Motors Institute, Flint, Michigan, December 11 and January 15; title: Industry Looks at Education.

W. K. Steinhagen, section engineer, Power Development Section, GM Engineering Staff, before members of Tau Beta Pi, University of Detroit, Detroit, Michigan, November 7; title: A Young Engineer Views the Automotive Industry.

Leonard E. A. Batz, head, Design and Standards Section, Special Design Engineering Department, AC Spark Plug Division, before the Shrine Club, Binghamton, New York, November 9; title: Engineering Trends; before the Boy Scouts' citizens now conference, Flint, Michigan, November 24; panel leader on Careers in Engineering; and before the Kearsley's School Men's Club, Flint, Michigan, January 7; title: Engineering Law and Ethics.

David C. Apps, head, Noise and Vibration Laboratory, GM Proving Ground, before the third west coast noise symposium, Los Angeles, California, November 13; title: Recent Developments in Traffic Noise Control.

Charles L. Tutt, Jr., administrative chairman, Fifth Year and Thesis program, General Motors Institute, before the Association of Technical Writers and Editors Society of Technical Writers, New York, New York, November 16; title: Copy for Industrial Reports.

Guy R. Cowing, president, General Motors Institute, before the Seventh Thomas Alva Edison Institute, West Orange, New Jersey, November 19; title: Science Education Possibilities in Cooperative Education; and before the Education Committee of the American Society of Mechanical Engineers, New York, New York, November 27; title: Distinguishing Characteristics of a Mechanical Engineer.

R. E. Tuttle, chairman, English and Cultural Studies Department, General Motors Institute, before the National Council of Teachers of English, St. Louis, Missouri, November 24; title: The Importance of Writing in the English Program.

W. R. Mackenzie, staff engineer, Product Information Department, Chevrolet Motor Division, before the Women's Advertising Club, Detroit, Michigan, November 26; title: Inside Chevrolet.

Robert J. Gleffe, plant personnel director, Central Foundry Division, before the Michigan State University regional foundry conference, Lansing, Michigan, November 30; presentation of student award for Michigan foundry conference; and before the St. Lawrence group, Canton, New York, December 6; title: What Industry Looks for in the College Graduate.

W. H. Pfeiffer, section head engineer, Materials and Process Laboratory, Frigidaire Division, before the Mid West Enamelers' Club, Chicago, Illinois, December 1; title: Fields of Application of Lower Temperature Porcelain Enamels.

Mauri Rose, special engine and vehicle development engineer, Product Information Department, Chevrolet Motor Division, before the students of Redford High School, Detroit, Michigan, December 3; title: Sticking-Your-Neck-Out Driving Belongs on the Race Track.

Loris H. Conrad, director, Production Engineering Department, Rochester Products Division, before the Irondequoit High School career day, Rochester, New York, December 5; title: Mechanical Engineering.

Dr. O. L. Crissey, administrative chairman, Personnel Evaluation Services, General Motors Institute, before the American Vocational Association conference, St. Louis, Missouri, December 6; title: Problems of Guidance and Counseling for Automotive Training.

H. O. Patterson, instructor, Psychology Department, General Motors Institute, before the southwest reading conference, Fort Worth, Texas, December 7; title: Reading Improvement Programs in Industry.

C. W. Lincoln, chief engineer, Product Engineering Department, Saginaw Steering Gear Division, before the American Society of Agricultural Engineers, Chicago, Illinois, December 10; title: Features and Applications of Ball Screws.

H. O. Haskitt, head, Speech Section, General Motors Institute, before the speech and theatre conference, Chicago, Illinois, December 27; title: Speech Programs for Present and Future Management.

Leon De Mause, administrative engineer, Chief Engineering Office, Cadillac Motor Car Division, before the Detroit Chapter of the American Society for Engineering Education, Detroit, Michigan, January 4; title: The Shortage of Engineers.

J. W. Bunn, Jr., representative, Plant Management Training Program, General Motors Institute, before the Junior Chamber of Commerce, Warren, Ohio, January 8; title: Creativity.

K. A. Stonex, assistant director, GM Proving Ground, before the Highway Research Board, Washington, D.C., January 10; title: Survey of Los Angeles Traffic Characteristics.

Industrial Engineering

A. J. Dunkle, director, Work Standards Department, AC Spark Plug Division, before the Whirlpool-Seeger Corporation management group, St. Joseph, Michigan, October 11; title: Philosophy and Practical Application of a Fair Day's Work Policy.

Robert W. Spinner, assistant chief engineer, Cleveland Ordnance Plant, Cadillac Motor Car Division, before the Industrial Mobilization Program, Ordnance Tank-Automotive Command, Centerline, Michigan, December 4; title: Industrial Mobilization as It Affects the

Vehicle Engineering Agency and Engineering Activities at Cleveland Ordnance Plant.

Leo J. Nartker, supervisor, Quality Control Department, Delco Products Division, before the Hamilton-Middletown subsection of the American Society for Quality Control, Hamilton, Ohio, January 9; title: Frequency Distribution.

Gerald H. Zimmer, general supervisor of packaging engineering, Receiving and Shipping Department, Delco Products Division, before editors representing trade magazines pertaining to packaging, New York, New York, January 23; title: Delco Products Division's Experience with Tri-Wall Corrugation in Production Packaging.

Claude M. Willis, safety director, Plant Safety Department, Delco Products Division, before the Dayton area Federal Safety Committee, Dayton, Ohio, January 25; title: Traffic Safety.

Manufacturing

Glen R. Fitzgerald, chief automotive engineer, AC Spark Plug Division, befor the A.S.M.E., New York, New York, November 29; title: Economic Considerations in Development and Use of Mechanical Assembly; and before the Industrial Fasteners Institute, Detroit, Michigan, January 10; title: Industrial and Mechanical Assembly Equipment.

Charles C. Brandon, works manager, Rochester Products Division, before the Rochester Materials Handling Society, Rochester, New York, December 12; title: Safe Handling of Material in Manufacturing Processes.

John F. Catalin, engineer-in-charge, Production Engineering Activity, Fisher Body Division, before the Cleveland Section of the American Welding Society, Cleveland, Ohio, December 12; title: Resistance Welding Applications in Automobile Body Production.

L. C. Goad, executive vice president of General Motors, before the S.A.E. annual meeting, Detroit, Michigan, January 16; title: Opportunities Unlimited for Engineers in Manufacturing.

G. R. Overman, chief experimental engineer, Product Engineering Department, Inland Manufacturing Division, before the Southern Ohio Rubber Group, Dayton, Ohio, February 6; title: Transfer and Injection Molding of Rubber.

Metallurgy and Bearing Materials

M. E. Otterbein, manager, Research and Development Department, Hyatt Bearings Division, before the Southwest Research Institute, San Antonio, Texas, November 6; title: Effect of Aircraft Gas Turbine Oils on Roller Bearing Fatigue Life; and before a group of engineers at Pratt and Whitney Division of United Aircraft Corporation, East Hartford, Connecticut, January 7; title: Effect of Aircraft Gas Turbine Oils on Roller Bearing Fatigue Life.

L. D. Cobb, manager of research and development, Product Engineering Department, New Departure Division, before the A.S.M.E., New York, New York, November 26; title: Discussion on High-Temperature, Heavily Loaded, High-Speed Test Rigs for Jet Engine Bearing.

C. W. Kalchthaler, chief engineer, Hyatt Bearings Division, before a group of engineers at Pratt and Whitney Division of United Aircraft Corporation, East Hartford, Connecticut, January 7; title: Precision Grade Bearings for Turbine Engines.

K. B. Valentine, assistant metallurgical engineer, Pontiac Motor Division before the S.A.E. annual meeting, Detroit, Michigan, January 15; title: Some Metallurgical Aspects of the Pontiac V-8 Engine Pearlitic Malleable Iron Crankshaft.

Research

Donald P. Koistinen and **R. E. Marburger,** research physicists, Physics and Instrumentation Department, GM Research Staff, before the Michigan Chapter of the American Chrystalographic Association, Detroit, Michigan, October 12; title: Effect of Depth of Penetration of X-rays on the Measurement of Quantities Having Large Gradients as a Function of Depth.

N. A. Hunstad, assistant head, and **M. C. Goodwin,** junior research technician, Fuels and Lubricants Department, GM Research Staff, before the National Lubricating Grease Institute, Chicago, Illinois, October 23; title: Effect of Vibration Frequency and Amplitude on Ball-Joint Steering Performance.

W. S. Coleman, assistant head, Gas Turbines Department, **M. H. Miller,** research physicist, Physics and Instrumentation Department, and **E. R. Mantel** metallurgical engineer, Metallurgy Department, GM Research Staff, before the Society of Experimental Stress Analysis, Columbus, Ohio, November 7; title: An Evaluation of the X-ray Diffraction Method of Stress Measurement with a Comparison to Dissection Methods of Residual Stress Measurement in Hardened Steel.

Leonard G. Johnson, senior research mathematician, Special Problems Department, GM Research Staff, before the A.S.M.E., New York, New York, November 25; title: The Use of High-Speed Computers in the Design and Appraisal of Helical Gears.

Arthur F. Underwood, manager, Research Staff Activities, and **Gregory Flynn, Jr.,** head, Mechanical Development Department, GM Research Staff, before the A.S.M.E., New York, New York, November 25; title: A Million-Pound, High-Speed, Dynamic Fatigue Test Machine.

R. A. Randall, senior research engineer, Fuels and Lubricants Department, GM Research Staff, before the Coordinating Research Council symposium, Chicago, Illinois, December 5; title: Some Problems of Reproducibility and Significance of Antiknock Ratings above 100 Octane.

C. F. Nixon, head, Electrochemistry Department, GM Research Staff, before the American Electroplaters Society, Mishawaka, Indiana, December 5; title: Evaluation of Durability; and before the A.E.S., Jackson, Michigan, January 8; title: Nickel Will Stretch.

C. A. Amann, supervisor, Gas Turbines Department, GM Research Staff, before the Engineering Society of Detroit, Detroit, Michigan, December 13; title: Turbines for Free Piston Engines.

Dr. F. E. Jaumot, Jr., director of research and engineering, Semiconductors Department, Delco Radio Division, before the American Association for the Advancement of Science and Sigma Phi Sigma, New York, New York, December 28; title: Modern Techniques in Diffusion Research.

Dr. L. R. Hafstad, vice president in charge of Research Staff, before the S.A.E. annual meeting, Detroit, Michigan, January 14; title: Basic Research and the Automotive Industry.

A Typical General Motors Institute Laboratory Problem:

Find the Horsepower to Give 4 Wheels of an Automobile a Certain Acceleration at a Specified Velocity

Faculty Member-in-Charge:
MERLE L. DeMOSS

G.M.I. Cooperative Students:
THEODORE N. LOUCKES
Oldsmobile Division
and ROBERT B. ROBINSON
Chevrolet Motor Division

When an automobile is accelerated from rest, a part of the available engine horsepower is absorbed in accelerating the 4 wheels. A certain portion of this absorbed power is used to increase the angular velocity of the wheels. The remainder is used to move, or translate, the wheels from their rest position. To calculate the horsepower required to accelerate the 4 wheels, the weight of the wheel-tire assembly, as well as the polar moment of inertia of each wheel, must be determined. One approach to this determination is the application of the principle of the torsional pendulum to obtain the polar moment of inertia.

A TYPICAL problem encountered in an analysis of automobile performance concerns determination of the horsepower required to accelerate certain portions of the car independently. A general understanding of the power required by each portion lends itself to an analysis of overall car requirements.

The analysis used to determine the horsepower required to accelerate the 4 wheels of a car differs from the usual procedure for determining the overall horsepower required to accelerate a car because of the problem of rotating the wheels, as well as translating, or moving, them down the road.

Before the horsepower requirements for accelerating the 4 wheels can be calculated, it is necessary to determine the polar moment of inertia of a tire and wheel assembly. A simple laboratory method to determine this value is to apply the principle of the torsional pendulum. In this method, the tire and wheel assembly is suspended at its axis of rotation by a steel wire (Fig. 1). The wire is fastened to a plate mounted centrally on the wheel. The wheel then is turned about its axis of rotation and the natural period of oscillation of the tire and wheel assembly plus the mounting plate observed. The same procedure can be used to obtain the natural period for the mounting plate alone.

Problem

A passenger car having 15-in. wheels was accelerated at a constant rate from rest to 60 mph in 10 seconds. The problem is to determine the maximum horsepower required to give *only* the 4 wheels this acceleration. Air and rolling resistance are to be neglected, and the assumption made that there was no slipping.

To obtain the following given data, the spare tire-and-wheel assembly was used for installation on the torsional pendulum. This was done to avoid any out-of-balance which might result from tire wear. Any 15-in. tire and wheel assembly could have been used, however, as long as it was balanced prior to installation on the pendulum. In actual practice the moment of inertia of a number of tire and wheel assemblies would be determined and the mean value used.

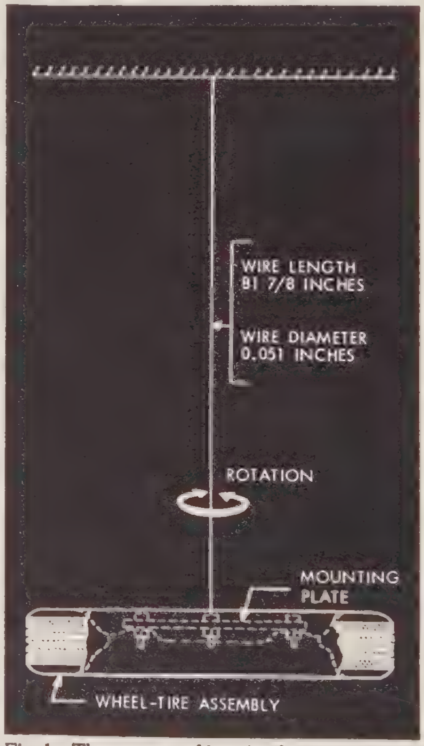


Fig. 1—The moment of inertia of a tire and wheel assembly can be determined by a torsional pendulum. The assembly is suspended from a mounting plate attached to a steel wire of known composition, diameter, and length. The wheel is turned about its axis of rotation, and the natural period of oscillation then observed.

Given Data

- Wheel revolutions per mile at 60 mph $= N = 723.35$
- Tire and wheel assembly:
  Weight $= W = 46.5$ lb
  Natural period of oscillation $= T = 61.81$ sec
- Pendulum suspension wire:
  Length $= L = 81\frac{7}{8}$ in.
  Diameter $= D = 0.051$ in.
  Modulus of rigidity $= G = 12 \times 10^6$ psi
- Mounting plate:
  Length = 6 in.
  Width = 6 in.
  Thickness = 0.25 in.
  Weight = 2.55 lb.

The solution to the problem will be presented in the July-August-September 1957 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

Presented here is a typical problem initiated and solved by students enrolled in advanced laboratory course work at General Motors Institute. This particular problem was solved during a 4-hour Automotive Chassis—Design and Testing Laboratory period. The problem allowed the students to apply principles of physics and dynamics to the solution of a typical automotive design analysis.

Contributors to Apr.-May-Jun. 1957 Issue of General Motors Engineering Journal

ROBERT M. BURTON,

co-contributor of the problem "Determine the Pattern Design for a Shell Molded Part from a Product Drawing" and the solution appearing in this issue, is a member of the Product Engineering Department of General Motors Institute, Flint, Michigan.

Mr. Burton joined the faculty of G.M.I. in 1953. In addition to his present duties as a teacher of engineering drawing and applied descriptive geometry, he also serves as co-ordinator of blueprint reading courses and as contact representative to the Spare Time Program for the G.M.I. Product Engineering Department. He assumed the latter duty in 1955. The blueprint reading courses are designed for training purchasing personnel, dealer cooperative trainees, apprentices, and spare time students. These courses include background for the interpretation of mechanical drawing and building construction.

As contact representative to the Spare Time Program, Mr. Burton's duties entail the coordination of special training programs which require courses in the Product Engineering Department area. He has charge of the procurement and supervision of G.M.I. Product Engineering Department part-time faculty for night school programs being conducted in Flint and Pontiac, Michigan.

Mr. Burton was granted the Bachelor of Science degree in 1947 from Northern Michigan College of Education. He received the master of arts degree in 1951 from the University of Michigan.

Mr. Burton is a member of the American Society for Engineering Education and the Michigan Industrial Education Society. In conjunction with his educational activities, he recently served on a committee to survey the needs of apprentices in their special and related fields.

WILLIAM C. EDMUNDSON,

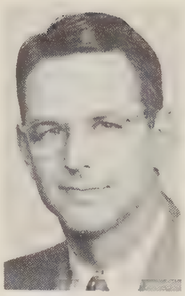
contributor of "A Discussion of Design Factors for a 12-Volt, 4-Pole, Wave-Wound Automotive Engine Cranking Motor," is a staff engineer in the Product Engineering Department of Delco-Remy Division. He currently is in charge of developmental work on high-output generators for application on air conditioned automobiles. The generators are designed to give high-output at high rotating speeds and high ambient temperatures. He also is in charge of starting motor and generator design and development for passenger car and farm tractor applications.

Mr. Edmundson has spent his entire engineering career with Delco-Remy. He joined this Division in 1934 after receiving the B.S.E.E. degree from Purdue University. Upon completion of a Delco-Remy student-engineer training program, he was assigned to the Product Engineering Department. In 1943 he became an assistant section engineer. He was promoted to section engineer in 1949, and assumed his present position a short time later.

Mr. Edmundson's previous major projects have included ignition equipment design and generator development work. During World War II, he was concerned with developmental work on generators for bomber aircraft application. Recently he was in charge of the design and development of 12-volt cranking motors and generators for V-8 high-compression engine electrical systems, of which he writes.

Mr. Edmundson is a member of Tau Beta Pi and Eta Kappa Nu, honorary societies. He is a member of the Society of Automotive Engineers and serves on the Society's V-Belt Subcommittee.

GEORGE A. HACH,



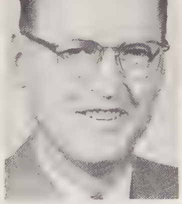
co-contributor of the problem "Determine the Pattern Design for a Shell Molded Part from a Product Drawing" and the solution appearing in this issue, is superintendent of the Process Engineering Department of Central Foundry Division's Danville, Illinois, plant.

This Department designs the special equipment necessary to produce grey iron and malleable iron castings by the green sand and shell mold processes. Product, pattern, and fixture design and development also are included under Mr. Hach's supervision.

Mr. Hach attended General Motors Institute from 1940 to 1942 as a co-op student sponsored by Central Foundry's Saginaw, Michigan, malleable iron plant. In 1944 he was granted the B.S.M.E. from the University of Michigan. He served with the United States Navy from 1944 to 1946.

On his return from military service he joined the Dow Chemical Company, later, the Dow-Corning Corporation. He returned to General Motors at Central Foundry Division as an engineer at the Saginaw plant in 1950. The following year Mr. Hach was recalled into the Navy. He returned to Central Foundry in February 1953 and was appointed as a customer contact in the Inspection Department. Eight months later he was appointed superintendent of the Process Engineering Department of the Danville plant. He served as superintendent of the Danville Malleable Foundry Department in 1956, returning to his present position as superintendent of the Process Engineering Department later that year.

GLENFORD M. HAVILAND,



contributor of "A Step in Body Manufacturing: Processing of Automotive Trim and Hardware for Production," is senior engineer-in-charge of the Trim and Hardware Styling Department of Fisher Body Division's Engineering Section.

Mr. Haviland joined Fisher Body's

Trim and Hardware Styling Department in Detroit as assistant engineer-in-charge in 1945 and was promoted to his present position in 1950. His paper describes the functions of this Department as it works in cooperation with the GM Styling Staff and the various GM car Divisions on the design and production of automotive trim and hardware.

Mr. Haviland's affiliation with General Motors started in 1931 as a General Motors Institute co-op student. Graduating from G.M.I. in 1935, he joined the Engineering Department of Pontiac Motor Division as a draftsman. In 1937 he transferred to the GM Styling Staff as a designer, returning to Pontiac Motor the next year with the assignment of working on advanced designs. In 1941 he was granted a leave of absence to work with North American Aviation Corporation in Los Angeles as an aircraft designer of surface controls. The following year he joined Fisher Body as a liaison engineer on aircraft, and was assigned to duties in Memphis, Tennessee. He was reassigned to Cleveland, Ohio, in 1943, where subsequent promotions led to his assuming charge of Fisher Body's liaison engineering office in Cleveland before transferring to Detroit to work with his present Department.

Mr. Haviland's technical society memberships include the Engineering Society of Detroit and the American Society of Body Engineers.

KENDALL O. KNIGHT,

co-contributor of "Methods Engineers at Work: Development of Semi-Automatic Assembly Equipment for Oldsmobile Front Suspension Control Arms," is a senior methods engineer in the Methods Engineering and Plant Layout Department of Oldsmobile Division.

Mr. Knight joined GM as a General Motors Institute co-op student sponsored by Oldsmobile in 1945. Under the G.M.I. cooperative program, his work assignments at Oldsmobile were in the Inspection, Final Assembly, Plant Engineering, and Methods Engineering Departments. After completing the G.M.I. 4-year program with a major in industrial engineering in 1949, Mr. Knight joined the Methods Engineering and Plant Layout Department. He was promoted to junior engineer in 1950. He received the B.I.E. degree from G.M.I. in 1951 and was promoted to methods engineer the following year. He assumed his present position in 1954.

In his capacity as a senior methods engineer, Mr. Knight is in charge of job improvements in Oldsmobile's press metal plant, engine plant, and paint plant. His work also entails assembly processing for front and rear axle assemblies.

Mr. Knight's previous major projects have included production planning for the J65 Sapphire Jet Engine compressor and turbine assembly manufactured by Oldsmobile between 1951 and 1954. He returned to automobile production when his assignment on the jet engine project was completed and contributed to the development of a machine which automatically polishes painted hoods and fenders. He also took part in production planning relating to the automotive engine plant expansion program.

WILLIAM L. LENNON,

co-contributor of "Methods Engineers at Work: Development of Semi-Automatic Assembly Equipment for Oldsmobile Front Suspension Control Arms," is a methods engineer in the Methods Engineering and Plant Layout Department at Oldsmobile Division. His current responsibilities include processing of front and rear suspension assemblies and initiating job improvements in Oldsmobile's axle and plating plants.

Mr. Lennon joined Oldsmobile as a junior methods engineer in the Methods Engineering and Plant Layout Department in 1952. He was promoted to his present position as methods engineer in 1954.

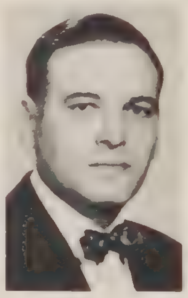
Mr. Lennon's previous projects with Oldsmobile have included the development of the use of leaf springs to hold parts on racks during plating operations, the development of a fixture to stake and drive lock pins simultaneously in the front suspension assembly, and the production planning for the new front suspension manufacturing facilities.

Mr. Lennon attended the University of Toledo, where he studied mechanical engineering, before transferring to Purdue University, which granted him the Bachelor of Science degree in mechanical engineering in 1952.

Before joining Oldsmobile Division, Mr. Lennon served with the United States Army from 1946 to 1947. After being separated from the Army, he was employed on various construction jobs in the Toledo, Ohio, area while studying for his degree.

In recognition of his scholastic work at Purdue University, Mr. Lennon was elected to the Pi Tau Sigma, honorary society.

Contributors' backgrounds vary greatly in detail but each has achieved a technical responsibility in the field in which he writes.



LEO MARCUS,

contributor of "A Mathematical Tool in Industry: An Algorithm for Curve Fitting by the Method of Least Squares," is a project engineer in the Engineering Calculation Department of Allison Division, which he joined in 1953.

Mr. Marcus' work in Engineering Calculation concerns mathematical and statistical analysis of problems involved in the research, development, and manufacturing activities of Allison. Included in this work is the programming of various engineering problems to be solved with the aid of electronic digital computers used by the Division. Another phase of his work is concerned with analyzing experimental data involving 2 or more independent variables and the determination of an analytical function to fit the observed data. To save time in this analysis, Mr. Marcus developed the method based on the principles of the method of least squares, of which he writes, to determine the minimum number of terms required in an analytical function to fit a set of observed data.

Purdue University granted Mr Marcus the Bachelor of Science degree in science in 1942 and the Master of Science degree in 1951. After graduation, Mr. Marcus

was employed as a physicist at the Naval Air Experimental Station, Philadelphia, where he helped to develop a portable piece of test equipment which was used for testing aircraft instruments without removing them from the airplane. From 1946 to 1953 he was an instructor of mathematics at the Purdue University Technical Institute. During this time he also was engaged in consulting work for the Department of Sanitation of the City of Indianapolis and the Cleveland Grain Company, Cleveland, Ohio.

Mr. Marcus is a member of Sigma Pi Sigma, honorary society.

LESTER MILLIKEN,

contributor of "The Cadillac Tubular Center X-Frame: A New Concept in Automotive Design," is a staff engineer in the Engineering Department of Cadillac Motor Car Division in Detroit.

Mr. Milliken is nearing the completion of 25 years' affiliation with General Motors, having joined the Engineering Department of Oldsmobile Division as a layout man in August 1932. His employment with Cadillac Motor Car dates back to April 1933, when he transferred to this Division as a layout man.

His long experience in the automotive engineering field has centered around chassis and frame design for passenger cars. As staff engineer, he is currently a member of Cadillac Motor Car's advanced design group which explores new possibilities in the passenger car field. Such considerations as the location of major components in an automobile, weight distribution, and cost trends are among those taken into account in exploring advanced designs, as well as the study of future developments in the car.

During World War II, Mr. Milliken was engaged in turret design and stowage for Cadillac tanks. From 1950 to 1955 he served as administrative engineer in charge of service activities for the Engineering Department.

Mr. Milliken is a member of the Engineering Society of Detroit and the Society of Automotive Engineers, serving as a member of the Passenger Car Activity Committee.

Before joining General Motors, Mr. Milliken was employed with the Locomobile Company of America, a pioneer automobile manufacturer, and the International Harvester Company, where he was engaged on a chassis design project for a proposed bus.

WALTER NOON,

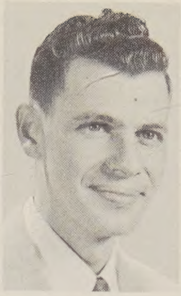
contributor of "Application of the Analog Computer to Engineering Problems," is a project engineer in the Electronics Department of the Process Development Staff, located at the General Motors Technical Center.

Mr. Noon is in charge of the Analog Computer Laboratory of the Electronics Department, and is responsible for the application of mathematics and simulation to the analysis and solution of the Staff's engineering problems. The analog computer, pictured in Mr. Noon's paper, has 30 operational amplifiers, 8 multipliers, 4 function generators, and an X-Y plotter available for computation, function generation, and recording. The problems on which he works arise in the design and development of mechanical, electronic, foundry, hydraulic, and pneumatic systems.

The Electronics Department of the Process Development Staff has a variety of equipment, including computing and ultrasonic devices, used in the development and testing of applied physics techniques for application to all phases of manufacturing processes and machine tool development. This equipment, including the electronic analog computer, is available for use by all GM Divisions, as well as the Process Development Staff.

Mr. Noon is a 1951 graduate of the University of Michigan, where he was granted the B.S. degree in physics. He received the M.S. degree in physics from Wayne State University in 1953, where he now teaches evening classes in physics.

Before joining the Process Development Staff in 1955, Mr. Noon was employed as a flight test analysis engineer with Lockheed Aircraft Corporation from 1951 to 1953. He also worked as a senior systems design engineer in electronics with Chance Vought Aircraft Corporation from 1953 until he joined General Motors.



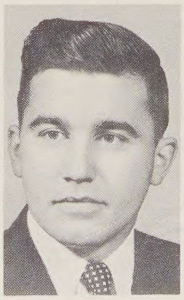
JOSEPH M. SHERWOOD,

co-contributor of "Industrial Engineers at Work: Some Typical Processing Problems Resulting from an Annual Model Change," is a senior process engineer in the Maintenance and Repair Department, Jigs and Fixtures, of Buick-Oldsmobile-Pontiac Assembly Division's Kansas City, Kansas, plant.

As a senior engineer, Mr. Sherwood is supervisor of the process engineering concerned with the tooling and processing relating to the assembly of Buick, Oldsmobile, and Pontiac automobiles. For the past 6 years, he has been concerned with the planning and execution of model changeovers for the Kansas City plant, as described in his paper.

One of 7 B.O.P. plants situated throughout the United States, the Kansas City plant was established shortly after World War II to assemble automobiles from parts produced in the factories of Buick, Oldsmobile, and Pontiac Divisions. At one time, the Kansas City plant assembled the Republic F-84F Thunderstreak concurrently with the 3 makes of cars. During this period the plant was capable of producing automobiles and aircraft simultaneously with the ability to convert to total production of either.

Mr. Sherwood joined B.O.P. Assembly Division as a General Motors Institute co-op student in 1947. At that time he was employed in the Personnel Department of the Kansas City plant. In March 1951 he became a junior engineer, and in October 1951 he was promoted to engineer. G.M.I. awarded him the B.I.E. degree in 1952. Continuing his work with B.O.P. Assembly, he was promoted to his present position in 1956.



ROBERT W. TRUXELL,

co-contributor of "Methods Engineers at Work: Development of Semi-Automatic Assembly Equipment for Oldsmobile Front Suspension Control Arms," is general supervisor of the Methods Engineering

and Plant Layout Department at Oldsmobile Division, Lansing, Michigan.

Mr. Truxell came to Oldsmobile as a General Motors Institute student in September 1942. His schooling was interrupted in 1943 by a 3-year tour of duty with the Army Air Force, after which he returned to Oldsmobile.

In June 1949 he graduated from General Motors Institute and began employment as a methods engineer. As such, he participated in new-plant planning and assembly processing for Oldsmobile's final-assembly plant. The following year, he earned the Bachelor of Industrial Engineering degree from the Institute through completion of a fifth-year project concerned with the development of a machine to set toe-in on conveyorized cars.

Additional projects included methods engineering planning for various new plants in connection with Oldsmobile's 90-mm cannon-assembly, jet-engine manufacturing, and bumper-plating activities. In May 1953 he was promoted to senior methods engineer and a few months later, to supervisor of methods engineering. He was promoted to his present position as general supervisor of the Methods Engineering and Plant Layout Department in 1955.

Mr. Truxell is a member of Alpha Tau Iota, honorary society, and serves on the GM Methods Engineering Subcommittee and the GM Master Mechanics Assembly Subcommittee.

ROBERT F. TUTTLE,

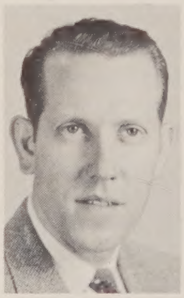
contributor of "Simple Tire Mounting Machine Keeps Production Lines Moving," is a senior process engineer in the Production and Standards Department of Chevrolet Motor Division. This Department is located at the Chevrolet Engineering Center, Detroit.

Mr. Tuttle joined the Production and Standards Department of Chevrolet in 1946 as a draftsman. He was promoted to senior process engineer in 1953.

Mr. Tuttle's current work is with the Tool Engineering Section of the Production and Standards Department, where he is engaged in the design and processing of tools and equipment for use in all Chevrolet assembly plants. It was in this capacity that he was responsible for the design and development of the tire mounting machine, of which he writes.

Some of Mr. Tuttle's previous major projects have included the development of a frame welder designed to weld body brackets to an automotive frame and a geometric steering machine. This machine has been in operation in Chevrolet assembly plants since 1955 and has set front wheel geometry on approximately 5,000,000 Chevrolet passenger cars. The operation of the geometric steering machine is to set caster, camber, and toe-in on the chassis before the car is completely built. This is accomplished by automatically loading the chassis to the weight of a finished car.

During World War II he served with the United States Army.



ROBERT R. WILLIAMS,

co-contributor of "Industrial Engineers at Work: Some Typical Processing Problems Resulting from an Annual Model Change," is a senior engineer—layout in the Buick-Oldsmobile-Pontiac Assembly Division Central Office, located in Detroit. His current assignment involves responsibility for the design and layout of proposed additions to B.O.P. assembly plant facilities.

Mr. Williams joined General Motors in March 1951 as a junior engineer in the Plant Engineering Department of the Kansas City, Kansas, B.O.P. assembly plant. Later that year he was promoted to engineer and in July 1952 became supervisor of plant layout. He was later transferred to the B.O.P. Central Office where he has worked as a senior methods analyst, in addition to his present position.

While at the Kansas City plant, Mr. Williams served as plant layout supervisor during the time F-84F Thunderstreak fighter aircraft were being produced simultaneously with Buicks, Oldsmobiles, and Pontiacs. He also was senior automotive methods engineer for this plant.

The University of Kansas granted Mr. Williams the Bachelor of Science degree in mechanical engineering in 1951. Before joining General Motors he was connected with the Wyandotte County, Kansas, engineering office.

Mr. Williams is a registered mechanical engineer in the state of Kansas. His technical affiliations include membership in the American Society of Mechanical Engineers.

ALFRED E. WILSON,

contributor of "The Importance of Keeping Records of Inventions" and this issue's "Notes About Inventions and Inventors," is a patent attorney with the General Motors Central Office Patent Section.

Mr. Wilson's work with the Patent Section is on special assignment, which includes the handling of negotiations pertaining to patent developments and the preparation of agreements relating to them. Both his educational background and professional experience combine the fields of patent law and engineering. He was graduated from the University of Florida in Gainesville with the LL.B. degree in 1931. In 1932 he was granted the A.B. and in 1933 the B.S.M.E. degrees from the same University. While working on his studies, Mr. Wilson taught aeronautical engineering at the University from 1930 to 1933.

He became a patent attorney with Bendix Aviation Corporation in South Bend, Indiana, in 1933, leaving in 1939 to become affiliated with the patent law firm of Dike, Calver, and Gray in Detroit. From 1942 to 1946, Mr. Wilson conducted his own law office in Detroit, specializing in patent and trade-mark practice. In 1946 he became assistant patent counsel for Packard Motor Car Company and continued with this organization as patent counsel from 1952 and after its reorganization as Studebaker-Packard Corporation. He joined General Motors in his present position with the Patent Section in July 1956.

Mr. Wilson is a member of the bar of the States of Florida and Michigan and several Federal District Courts. Among his professional society memberships are the American Patent Law Association, the Patent Section of the American Bar Association, and the Michigan Patent Law Association. He served as president of the last Association from 1955 to 1956.

ENGINEERING

ASSIGNMENT IN GM

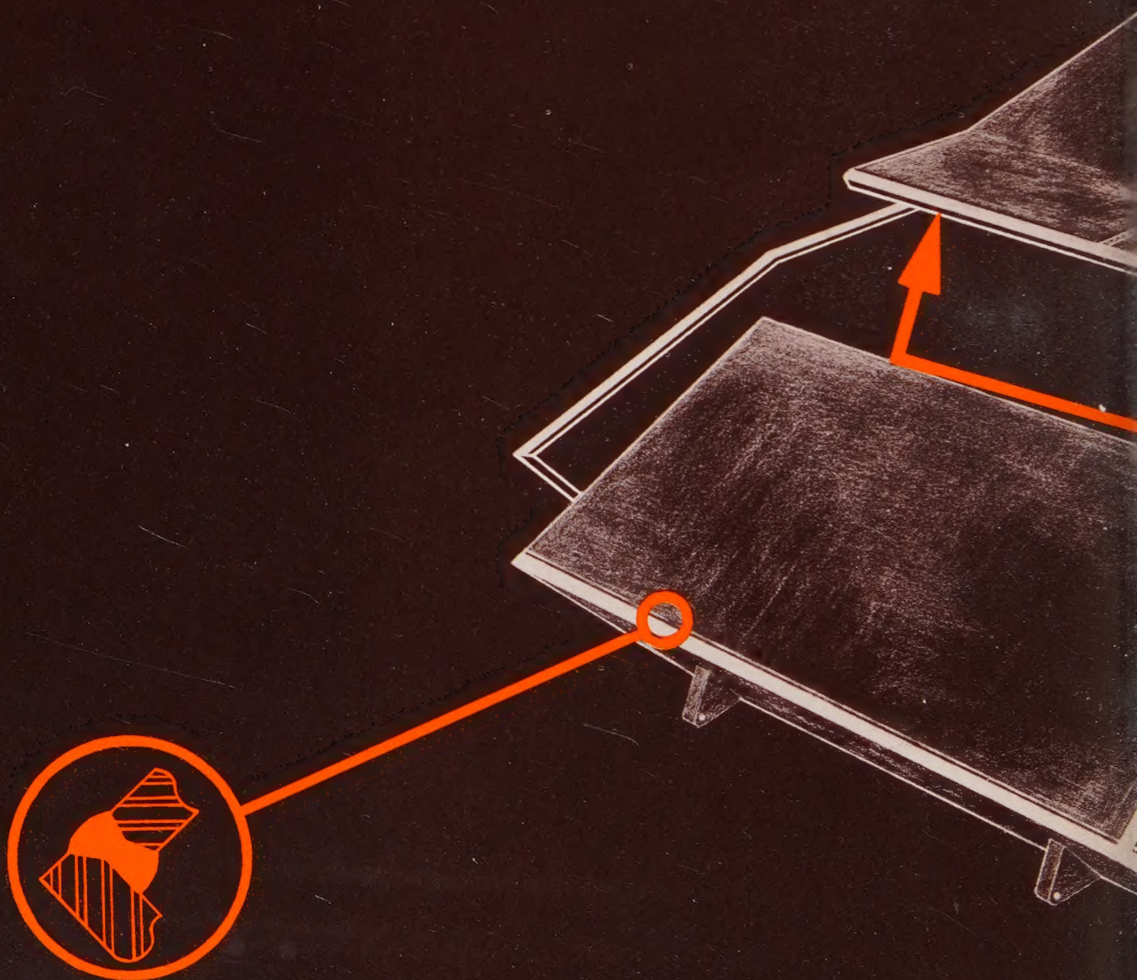
Diesel engine performance and economy are affected directly by the characteristics of the fuel spray. These spray characteristics, in turn, are determined by the injection nozzle and other components in the fuel system. To gain information on the effect of various engine designs on fuel spray characteristics, personnel of the GM Engineering Staff conduct analytical tests of Diesel fuel system components at the General Motors Technical Center.

Edwin J. Piersma, an experimental engineer in the Power Development Section of the Engineering Staff, is shown here conducting an analysis of experimental Diesel fuel system components by stroboscopic study of spray characteristics and injection timing. This test apparatus includes a complete fuel system to duplicate conditions in an actual engine. Visual observation is made by adjusting the stroboscopic light to the frequency of the fuel injection spray. By regulating the light frequency, the engineer can observe the spray at various stages in its development. In the accompanying photograph, Mr. Piersma is noting the successive wave frequencies of the spray as they appear on the screen before him.

Such analytical studies provide information on the effects which the characteristics of the fuel spray have on Diesel engine performance and fuel economy. In addition, the characteristics of the spray provide valuable information on such variables as nozzle design, fuel line dimensions, pump delivery valve retraction, plunger size, and cam velocity.

Mr. Piersma has been an experimental engineer in the Power Development Section since January 1954. His work includes conducting dynamometer studies and bench tests of engine components, as well as fuel system evaluation. Mr. Piersma is a 1951 graduate of the University of Michigan, where he was granted the Bachelor of Science degree in mechanical engineering. Upon graduation, he joined General Motors as a graduate engineer-in-training with the Engineering Staff. In 1953 he was promoted to junior engineer before assuming his present position.

GENERAL MOTORS ENGINEERING JOURNAL



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